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PAPERS PRESENTED

AT THE

ELASTOMERIC BEARING SYMPOSIUM

18 MARCH 1969

US ARMY AVIATION MATERIEL LABORATORIES

FORT EUSTIS, VIRGINIA

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106

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FOREWORD

Contained herein is a compilation of papers presented at the Elastomeric Bearing Symposium held at the U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, 18 March 1969.

The objective of this symposium was to disseminate information generated to date on elastomeric bearings to aircraft airframe manufacturers as well as cognizant Government personnel.

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INTRODUCTION

by

J. N. Daniel
U.S. Army Aviation Materiel Laboratories

The introduction of the helicopter into military as well as commercial applications brought with it a bucketful of problems that had not been encountered in fixed-wing aircraft experience. Many of these problems have been solved, or at least reduced drastically, but the problem of adequately supporting highly loaded oscillating mechanisms continues to plague us.

Initially, conventional rolling contact bearings were used with a drastic derating to compensate for the reduced lubrication. Lubricants were changed from grease to oil. Tension torsion straps were used to unload the pitch change bearings. New materials such as Teflon and Fabroid were introduced. Hingeless hubs removed some of the oscillating joints, etc. All of these developments made the situation better. In fact, the degree of improvement was so great that some of us may have been lulled into a complacent state.

Our experience in the extremely adverse environment in Vietnam has demonstrated that complacency is not a good state to be in. Bearing, seal, and lubricant problems and excessive maintenance requirements have been the order of the day.

I would like to quote a portion of a report made less than a month ago by a member of this Command who is serving as Aviation Consultant to the Science Adviser of the Military Assistance Command, Vietnam - or MACV. This report was made after a trip to the 1st Air Cavalry Division located at Phuoc Vinh, where the CH-47, UH-1, and AH-1 helicopters are used.

> "Rotor system problems are constantly encountered on all of the helicopters. It was reported on the 47's that the hinge pin seals leak badly (on 14 of 17 aircraft in one unit), that they are experiencing excessive wear of pitch change link rod end bearings (required to pull every 100 hours for bench check).

"The UH-1's are experiencing bearing problems in the main and tail rotors and swashplate area, reportedly caused primarily by hardening of the old grease, making it difficult to properly lubricate. The 540 head on the AH-1G Cobra is giving many problems, with the Teflon bearings lasting only 300-350 hours. Trunnion caps are overheating and the bearings are being chewed up. The grips are also rapidly wearing out, and the main rotor pitch change link rod end bearings are sometimes going bad after only 150 hours (the bearings have a scheduled TBO greater than 3000 hours). The Cobra's tail rotor hub still requires inspection every 100 hours, although the throwaway rate has been greatly reduced due to relaxing the inspection requirements. Bearings in the tail rotor assembly are still a problem, however, and hardening of the old lubricant is believed to be the primary cause. All of these items contribute to the many hours of unscheduled maintenance that are being performed."

This is not intended to point a finger at any particular aircraft or manufacturer, but only to indicate that nonrotating bearings are still causing us major concern.

One of the many definitions of symposium is "a conference at which a particular topic is discussed by various speakers." As a result of the discussion in this symposium, we hope to accomplish two important objectives. The first is to promote the exchange of information between the investigators and the potential users. This exchange should include a free discussion as to what, exactly, the various test results mean. The second objective is to stimulate each one of us to devise some workable solutions to the problems which confront us in the current operation of military helicopters. These problems will continue to plague us in the operations of future developments of helicopters unless we can come up with some solutions now.

I certainly don't mean to imply that elastomeric bearings will be a panacea for all our troubles, but if we can get rid of each troublesome area, then we will be making real progress toward the development of highly reliable aircraft. MG John Norton was speaking to the producers of Army aircraft recently, and he said:

"The current basic goals are to develop new aircraft that are more maintainable, more reliable, more survivable, and less vulnerable than those now in service....I specifically challenge each member of the research and development community of Government and industry to dedicate every skill available to him to assist us in striving to reach our goals."

It was our intention, when planning this meeting, that we would have Mr. Emil Thomas of the Air Force review the work that has formed the basis of most of the subsequent efforts that have been made toward the development of elastomeric, laminated bearings. Unfortunately, Emil's office had used up all their available travel funds, so he was unable to be with us. He did dig out a lot of historical information from his files, and I will go over some of this now.

In April of 1955, Mr. W. L. Hinks made an application for a patent on a new type of bearing for use where only oscillating motion is required. In mid-1956, Mr. Hinks was corresponding with Mr. Hank Velkoff of the Air Force at Wright-Patterson Air Force Base on the applicability of this type of bearing in helicopters. This led into a test program which was sponsored by TRECOM, our predecessor organization, and conducted by the Air Force with Randolph Research Co. (Hinks' organization).

Figure 1 shows the fatigue test machine used during the test program. Two bearings are installed back-to-back in the machine. Axial load is applied by torquing-down the nut visible in the photograph. An aluminum washer, instrumented with strain gages, was used to measure the applied axial load. Oscillatory motion is applied through the lever arm. Bearing temperatures and oscillation speed were monitored. Figure 2 shows the instrumentation setup. Figure 3 shows one of the bearings that was constructed of brass laminations bonded together with a natural rubber elastomer.

Some of the results of this test program are shown in the next figures. Figure 4 is a plot of static deflection versus axial load. It can be seen that the variation is essentially linear throughout the range of test conditions. It is also noteworthy that these bearings, which were manufactured in a backyard type of facility, with little control of uniformity, were so nearly alike.

Figure 5 shows the results of the torque/deflection test at various axial loads. This again shows a good linearity and also an insensitivity to axial load.

Figure 6 shows the static torque versus ambient temperature for various angular deflections. This shows that the natural rubber is not very flexible at low temperatures.

Figure 7 shows another way to look at the temperature effect on natural rubber. This shows the temperature of the bearing as a function of operating time in a number of ambient temperature conditions. It should

be noted that it does not take very long to raise the temperature above 0°, even in a -40° ambient condition.

In fatigue tests, one set of bearings was operated for 1065 hours at 300 cycles per minute, ± 5° twist angle, and 20,000 pounds compressive force. No distress was noted. The twist was increased to ± 7.5° and an additional 282 hours of operation before failure. Additional tests were conducted with similar results. A failed bearing is shown in Figures 8 and 9.

These tests were conducted in 1960 and 1961. Later on today you will probably see the results of tests that have been conducted since then on bearings like these and also on various other schemes which have been devised to accomplish the same objectives.

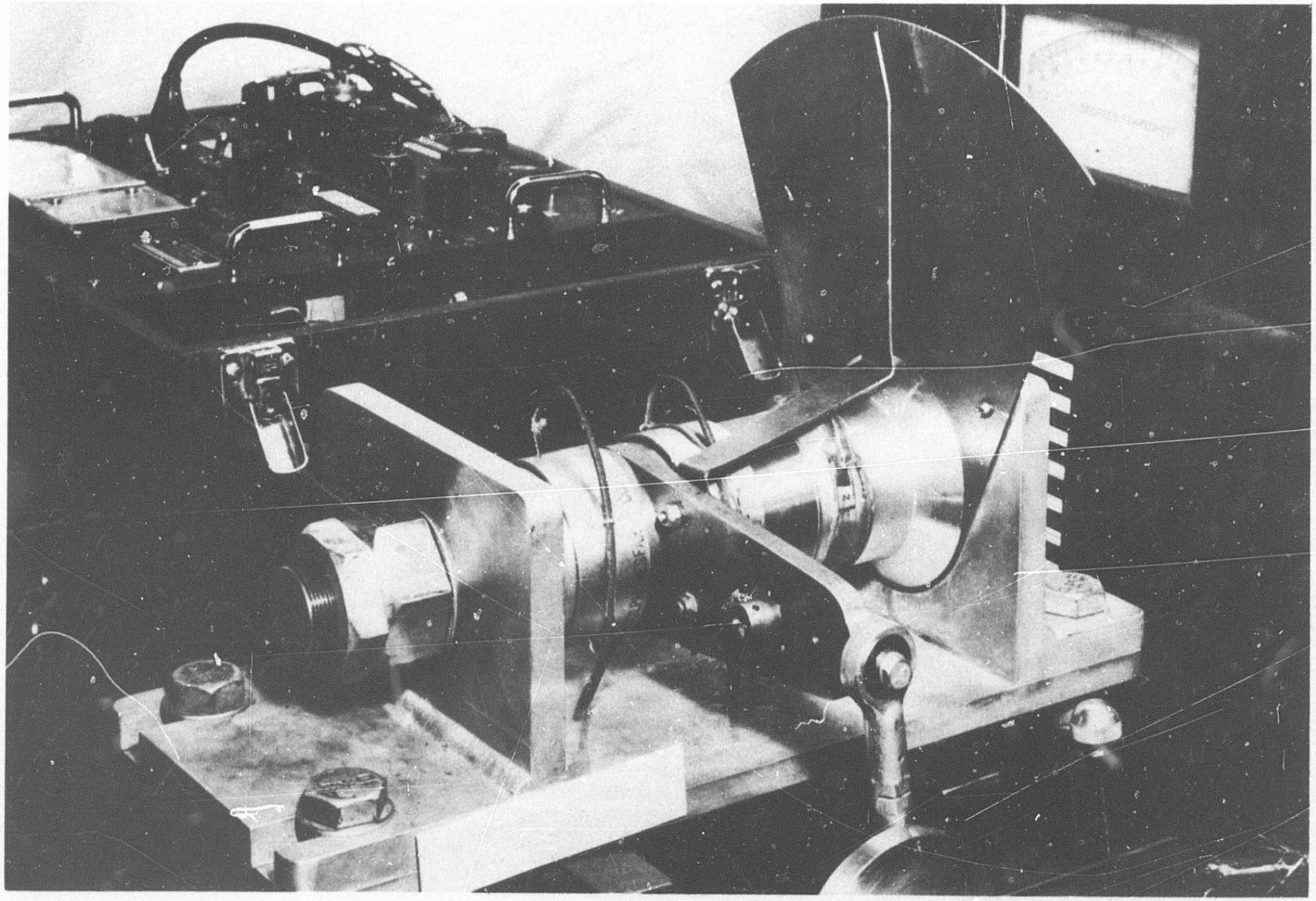

Figure 1. Laminated Pad Bearing, Back-to-Back Test Rig, With Two Mounted Bearings.

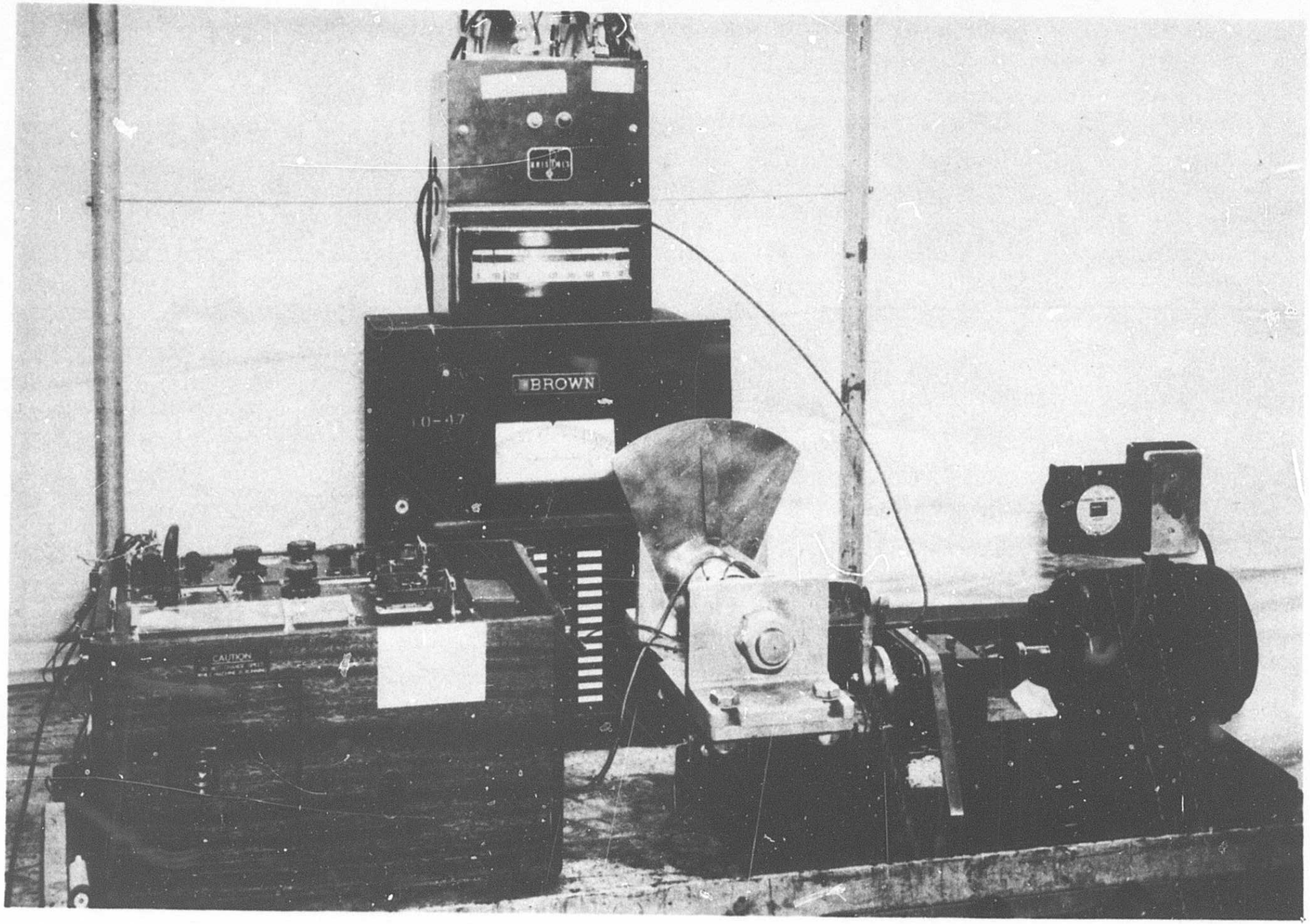

Figure 2. Laminated Pad Bearing Endurance Test Rig and Instrumentation.

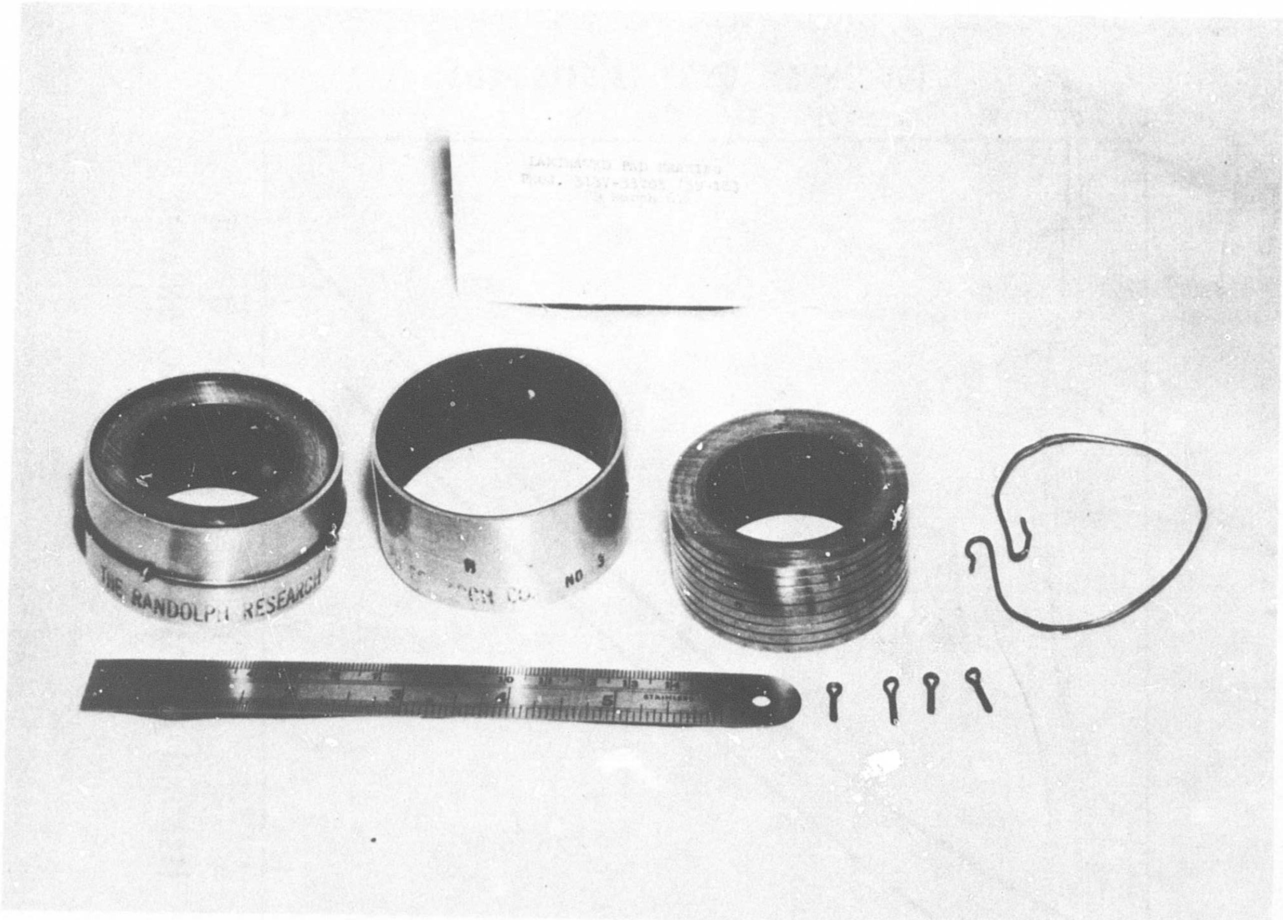

Figure 3. Laminated Pad Bearing.

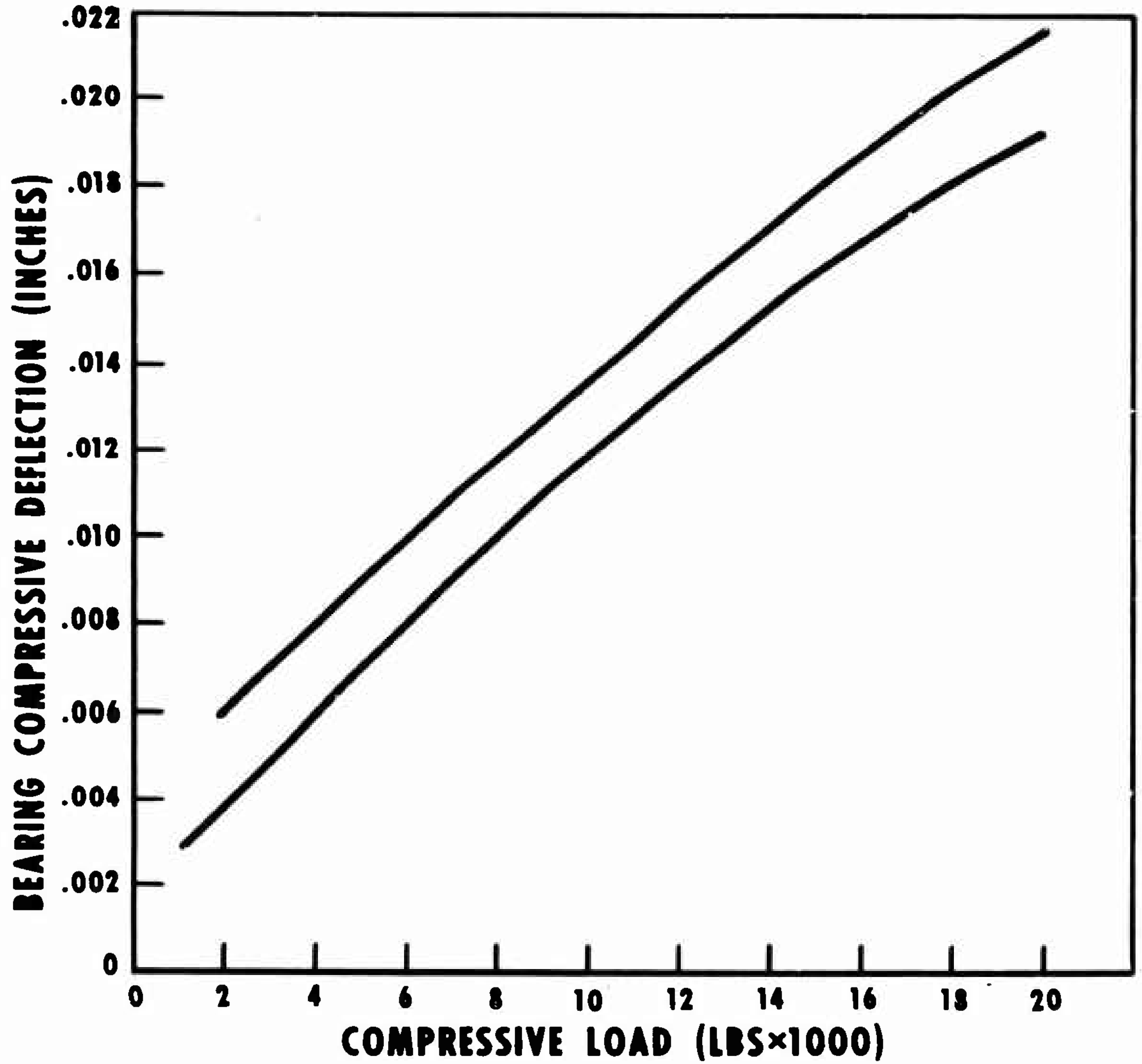

Figure 4. Static Deflection Versus Axial Load.

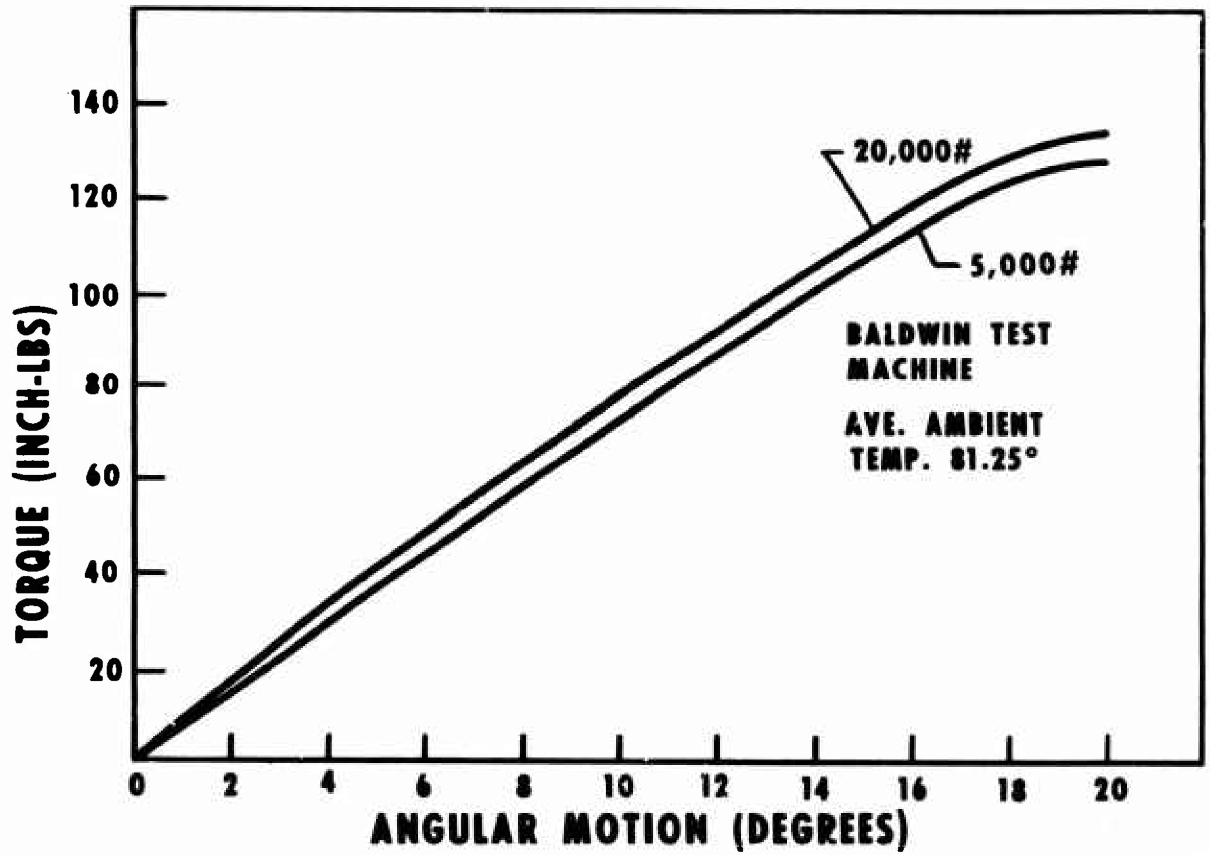

Figure 5. Torque/Deflection Static Test.

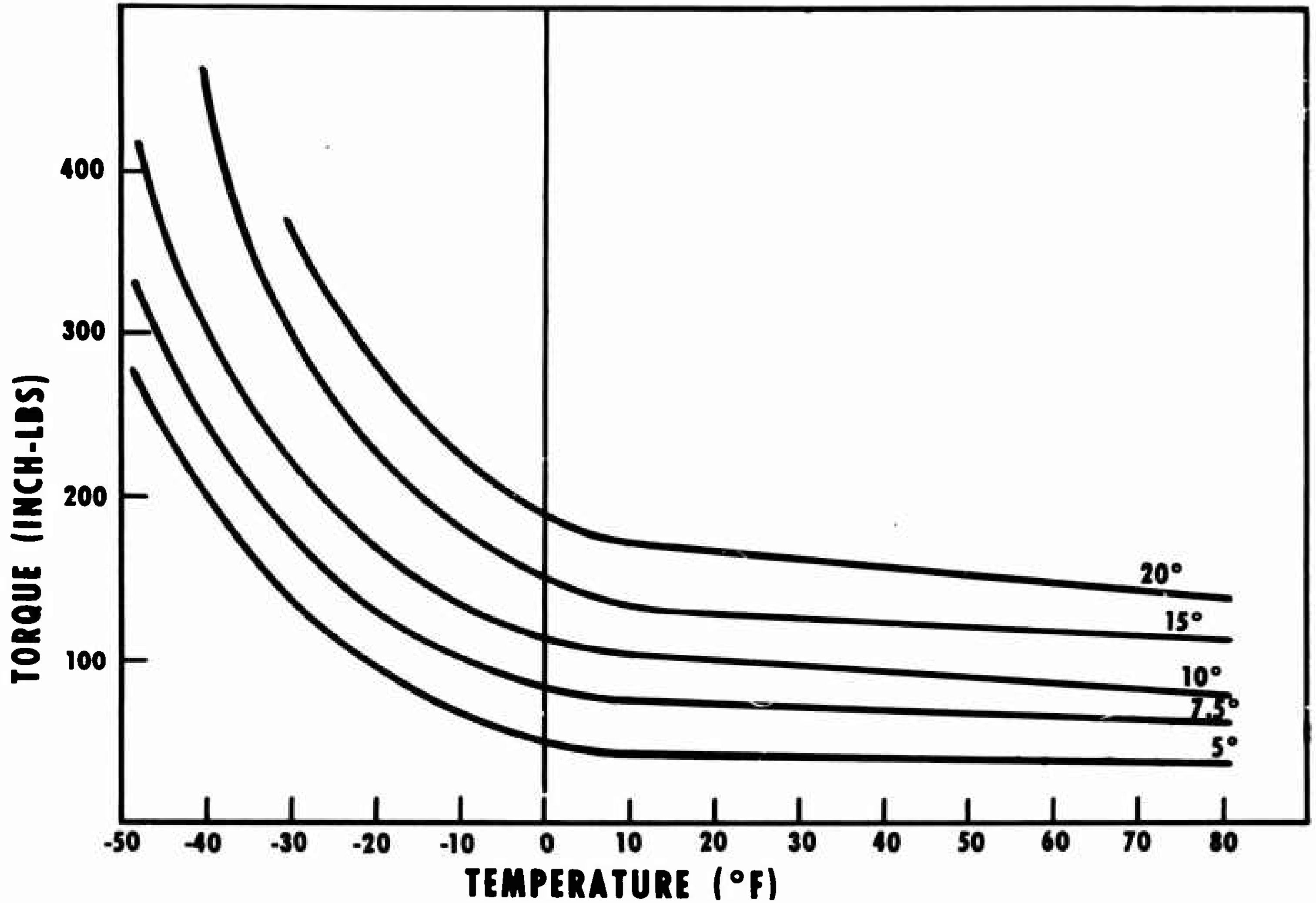

Figure 6. Single Bearing Torque for Angles of Twist Versus Temperature.

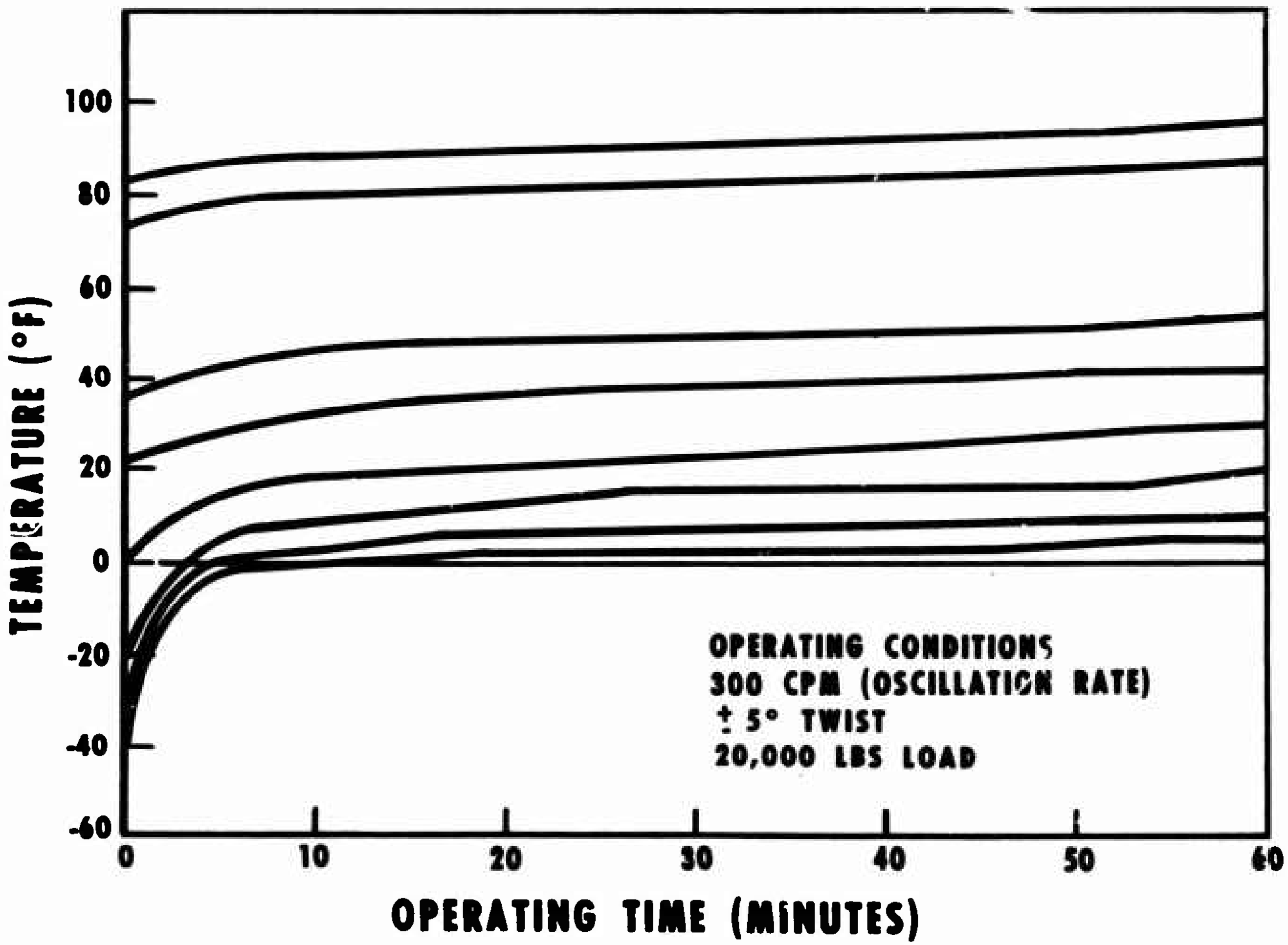

Figure 7. Temperature Surveys.

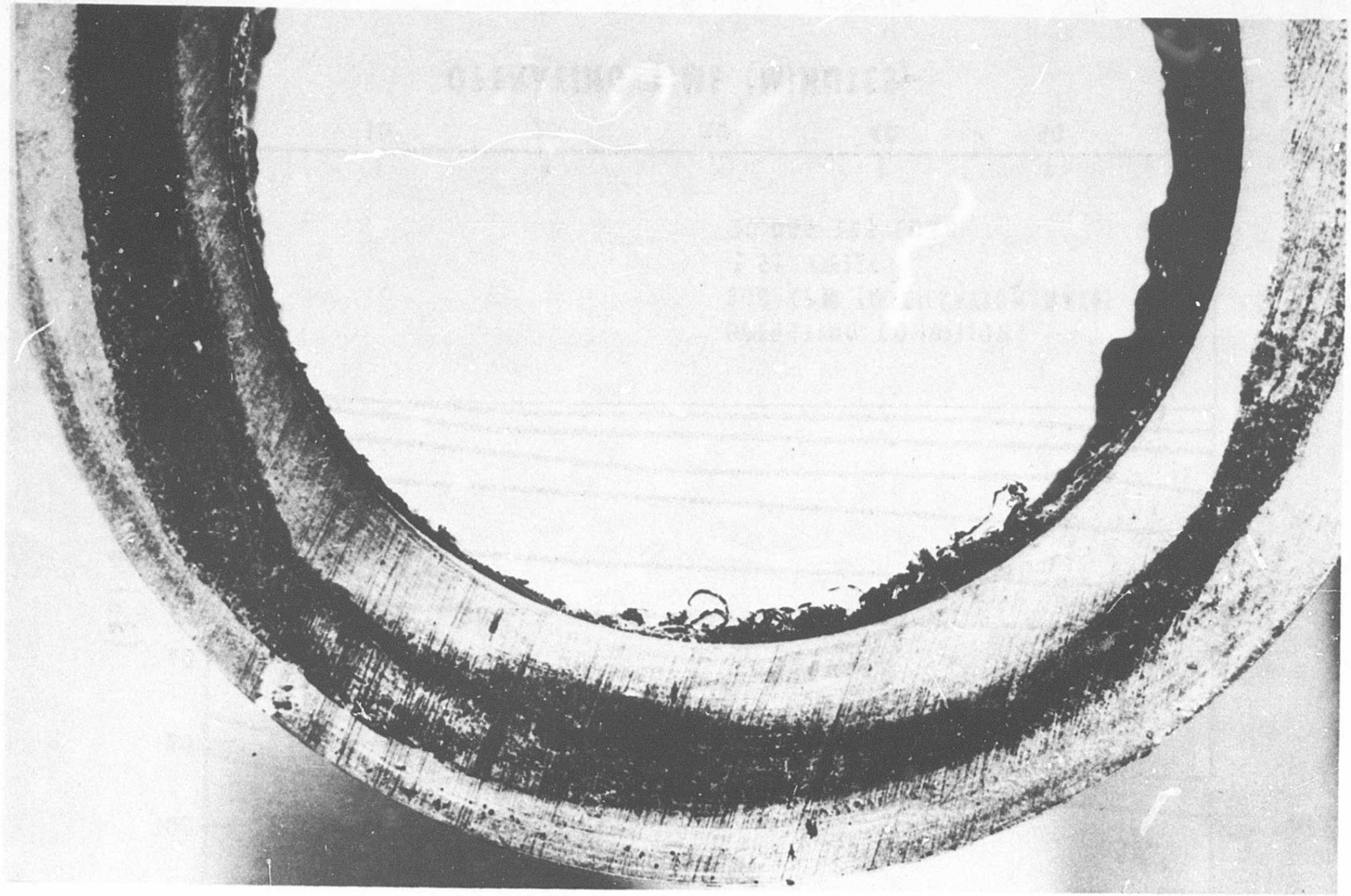

Figure 8. Laminated Pad Bearing (Failed After Equivalent to 671:28 Hours Endurance).

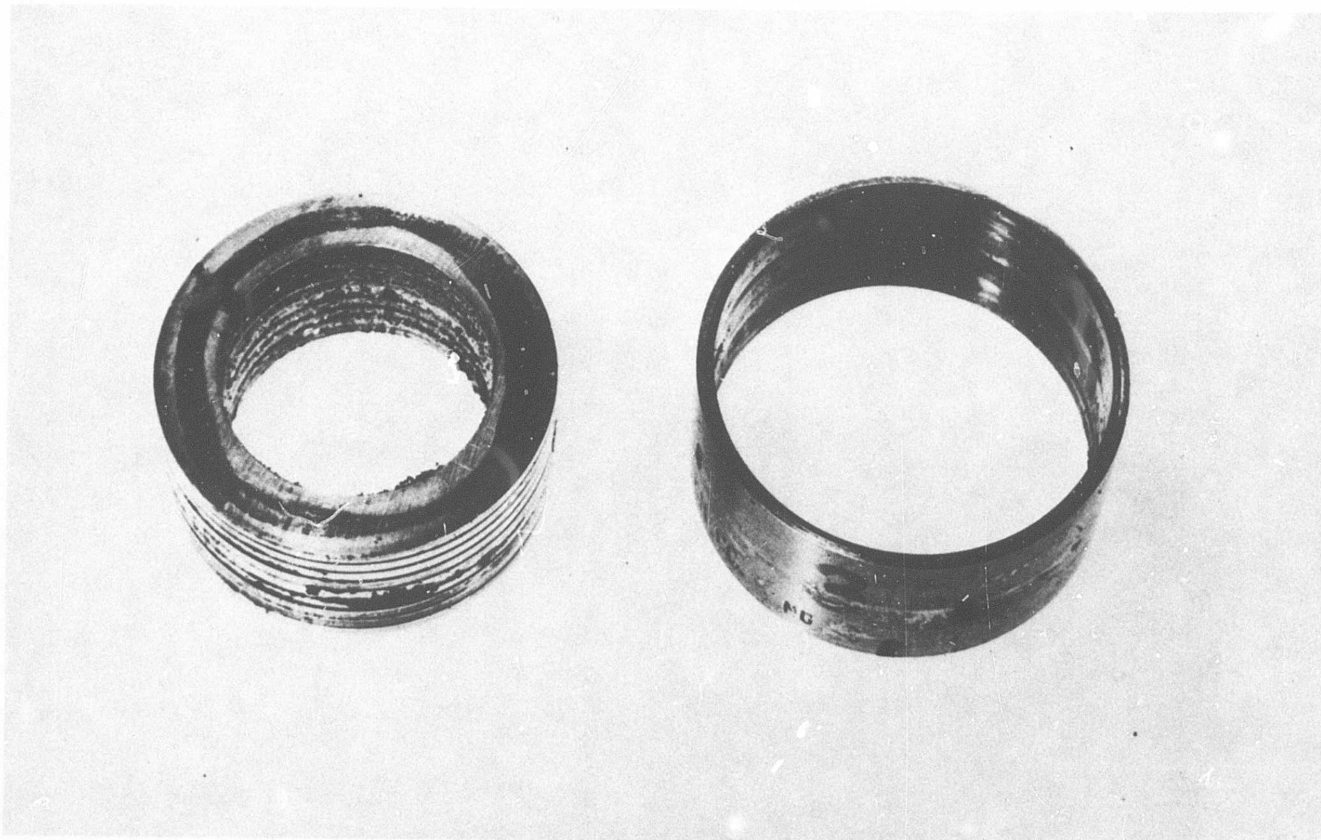

Figure 9. Laminated Pad Bearings (Failed After Equivalent to 671:28 Hours Endurance).

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ELASTOMERIC LAMINAR BEARING ANALYSIS

by

Mr. Z. Zudans
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Standard analysis methods, when applied to elastomeric bearings, give rise to significant difficulties. These difficulties are associated with the facts that:

1. Elastomer is incompressible (i.e., Poisson's ratio $\nu = 0.5$).

2. Actual strains in elastomeric bearings are large (20% to 200% are not unusual).

3. Stress-strain curve is nonlinear, but for the most part elastic, with rather small hysteresis effects.

4. Engineering properties of elastomer are not well defined.

5. Design of elastomeric bearings is such that the layers are very thin compared to the overall dimensions of the bearing.

For an incompressible material (Poisson's ratio $\nu = 0.5$), the classical elastic field equations, including Navier's equations of equilibrium, are not valid. As a consequence, when Poisson's ratio approaches 0.5, the solution of these equations by numerical techniques results in large errors. The reason for this behavior can be simply illustrated by looking at the equilibrium equations of the classical linear theory of elasticity.

$$G\nabla^2 u_i + (\lambda + G) e_{,i} + X_i = \rho \frac{\partial^2 u_i}{\partial t^2} \tag{1}$$

where G is the shear modulus and

$$\lambda = \frac{\nu E}{(1+\nu)(1-2u)} \tag{2}$$

both known as Lamé's constants. When $\nu \to 0.5$, from Equation 2 we note that $\lambda \to \infty$ unusable in the above form.

Equation 1 and other field equations of the linear elasticity have been reformulated by Herrmann and Toms[1] so that they are valid for any admissible Poisson's ratio, including 0.5. Similarly, Herrmann[2] has derived a variational theorem which allows us to handle the same range of the Poisson's ratio.

Using this modified definition, the linear small deformation case can be handled quite readily by numerical techniques such as the finite element method. Solution of this type, however, due to the large actual strains resulting in the elastomeric bearings, is of no practical significance. For proper accounting of large displacements, one must use the nonlinear strain-displacement relations. Similarly, the physical nonlinearities as evidenced by the stress-strain relations must be taken into account when analyzing the elastomeric bearings.

The best way to handle the geometric and physical nonlinearities is by the advanced techniques of the incremental analysis.[3] Incremental analysis is derived from the Incremental Principle of Virtual Work[3]

$$\int_\tau \left(S_{ij}\ \delta\eta_{ij} + \Delta\sigma_{ij}\ \delta e_{ij}\right) = \int_\tau \underset{\sim}{F}.\delta\underset{\sim}{u}d\tau + \int_\sigma T.\sigma\underset{\sim}{u}d\sigma \tag{3}$$

which represents the relations governing small incremental displacements δu in the body subjected to the initial accumulated stresses S_{ij}. By using Equation 3, a finite element method of solution can be devised such that the accumulation of incremental steps of loading results in a large final deformation. Also, by applying a tangent modulus method to the stress-strain curve, the nonlinear stress-strain curve can be followed elastically or along a known hysteresis loop during the unloading.

A sample problem, Figure 1, was solved by using Equation 3 in a special computer program developed at The Franklin Institute Research Laboratories. Solid lines indicate the deformed shape after straining the body to 10% strains, approximately; dashed lines indicate the original undeformed shape. This rubber washer was compressed to a dished top surface as indicated in Figure 1. Figures 2 and 3 indicate the maximum and minimum principal stresses in a radial plane. Figure 4 gives the stress-strain curve used for this solution. This curve can be symbolically expressed by

$$\sigma = \sigma(\varepsilon) \tag{4}$$

where σ and ε can be related to the actual stress σ_{ij} and strain e_{ij} tensors by

$$\sigma = \sqrt{3(1/2\, c_{ij}\, s_{ij})}$$

$$\varepsilon = \frac{1}{1+\nu} \sqrt{3(1/2\, e_{ij}\, e_{ij})} \tag{5}$$

where s_{ij} is the stress deviator

$$s_{ij} = \sigma_{ij} - 1/3\, \sigma_{kk} \tag{6}$$

The computer program based on Equation 3 can be used to solve the elastomeric bearing problems in general. This solution technique is well developed, and the only limiting factor on its wide application is its economy. For general structural problems, such as a solid rubber washer shown in Figure 1, the representation by the discrete network of finite elements is simple and economical. If one considers the actual geometry of the elastomeric bearing, one notes that the layers, both metallic and rubber, are thin and numerous. Proper representation by finite elements requires these elements to be of approximately square shape. This, however, would lead to extremely large numbers of elements, which in turn would dictate long computer running times, in particular, because of the incremental character of the solution. Accordingly, while a good method of solution is available the peculiar geometry of the elastomeric bearings makes its application unattractive from the economical point of view.

In addition to the effects of nonlinearities described above, there remains the question of the availability of the engineering properties of rubber. From the literature available on the subject, it appears that there is a great need for experimental determination of a reliable engineering property of various elastomeric compounds.

Economics of the finite element analysis approach can be considerably improved by devising a special finite element consistent with the geometry of the particular elastomeric bearing construction. The specific geometric constraint of layer thinness and multiplicity can be used by setting up a multilayer solid model and by constraining the layers to no-slip, no-delamination condition at their interfaces. The resulting finite elements would consist of three-dimensional curved sandwich surfaces consisting of a metallic and an elastomer layer. This element would allow the handling of large bearing systems with fewer elements and a more economical solution. A similar principle is used in the shell finite element

analysis, except that in the present case there would be multiple elements through the entire bearing thickness.

It is my opinion that such a specialized version of the finite element computer analysis would represent a practical design tool and, hence, that it should be developed.

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1. Herrmann, L.R., Toms, R.M., A REFORMULATION OF THE ELASTIC FIELD EQUATION, IN TERMS OF DISPLACEMENTS, VALID FOR ALL ADMISSIBLE VALUES OF POISSON'S RATIO, J. Appl. Mech., Vol. 31, pp. 148-149(1964).

2. Herrmann, L.R., ELASTICITY EQUATIONS FOR INCOMPRESSIBLE AND NEARLY INCOMPRESSIBLE MATERIALS BY A VARIATIONAL THEOREM, AIAA Journal, Vol. 3, No. 10, pp. 1896-1990 (1965).

3. Biot, M.A., MECHANICS OF INCREMENTAL DEFORMATION, J. Wiley, N.Y. (1965).

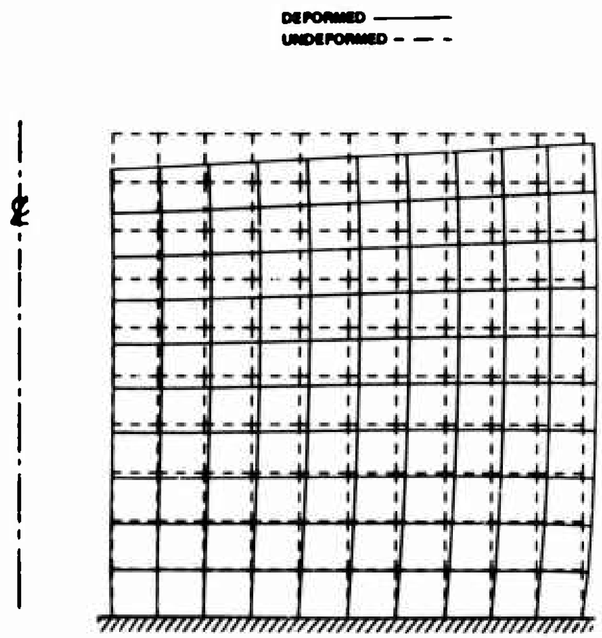

Figure 1. Rubber Washer Compressed Between Flat Conical Surfaces.

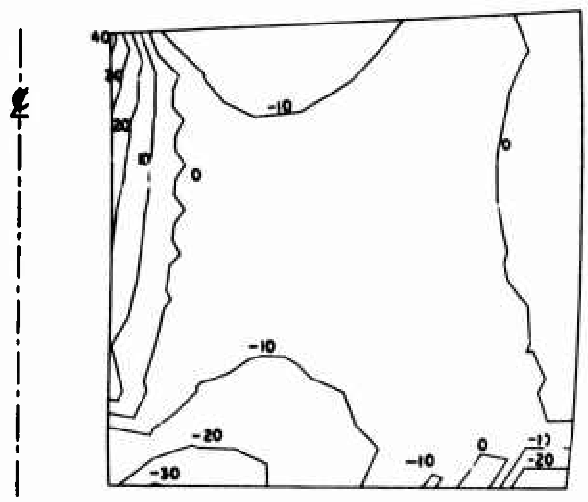

Figure 2. Maximum Principal Stress, Lb/In.^{2}.

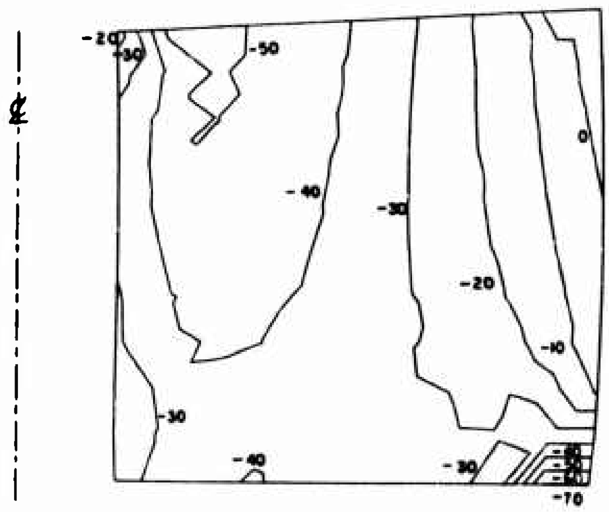

Figure 3. Minimum Principal Stress, Lb/In.^{2}.

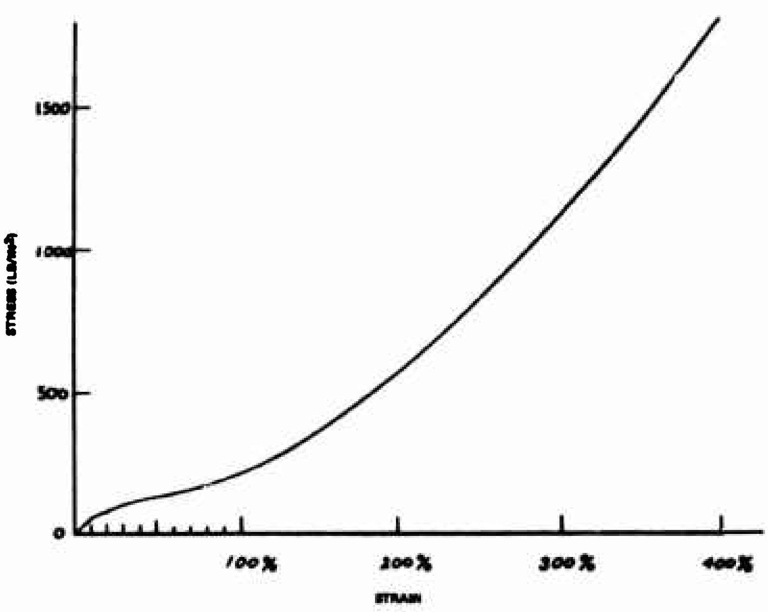

Figure 4. Stress-Strain Curve for Rubber-Like Material (Ref: McPhereson Klemin Engineering Uses of Rubber, p. 64).

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DESIGN AND DEVELOPMENT OF AN ELASTOMERIC-BEARING ROTOR HUB

by

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SUMMARY

The need for a simplified, more efficient, and fail-safe helicopter rotor system is of prime importance to the future of the helicopter industry. An approach to such a simplified, articulated rotor hub designed for advanced-geometry blades, such as glass-fiber-reinforced composite or boron blades, is presented. Design considerations and development tests are described for a rotor hub that provides for rotor blade flap, lag, and feathering freedom by means of a single, universal bearing of a unique and fail-safe design, called the elastomeric bearing, rather than by three separate hinges. The development program includes building and whirl testing a full-scale elastomeric-bearing rotor hub for a growth version of the CH-47 helicopter.

The elastomeric bearing consists of concentric spherical laminations of an elastomer and metal bonded together in thin alternate layers. The bearing is mounted so that the blade centrifugal force results in compression of the elastomer, while pitch, lag, and flap motions result in shear deflection of the elastomer. This design, as demonstrated by bench tests, results in a fail-safe feature of the bearing, which even in a case of complete deterioration of the elastomer excludes the possibility of blade separation from the rotor hub.

Achievement of fail safety of the hub proper requires further studies, but appears feasible because the simple geometrical shape permits use of such crack indicating devices as crack detection wires or pressure loss indicators.

The elastomeric-bearing rotor hub maintains all the advantages of an articulated blade retention system in a greatly simplified rotor hub while providing the following:

1. Reduced maintenance and cost, improved reliability, and fail safety

2. Improved control sensitivity by increased rolling moment and roll damping

3. Better blade/fuselage clearances

4. Reduced weight

5. Increased airspeed within the specified vibration level

6. Compatibility with the advanced-geometry glass-fiber-reinforced plastic rotor blade and the advanced fibrous-reinforced composite (boron filament) rotor blade

Rotor head conceptual layouts using spherical elastomeric bearings showed the possibility of simplifying articulated hub design as well as achieving drag, weight, and cost savings. Detail design studies resulted in satisfactory solutions of such key design problems as:

1. Permanent droop and flap stops

2. Centrifugal droop and flap restrainers

3. Pitch locks

4. Blade attachment and manual folding

5. Adaptation of power blade folding

6. Blade lag damping

7. Rotor hub fairing

To permit determination of efficient bearing design for a given helicopter blade motion and an allowable elastomer strain, a method for theoretical analysis was developed and converted into a computer program for a spherical elastomeric bearing by which the strain in the rubber layers for either pitch or flap deflection could be determined. Also, the combined strain at any point in any layer for a given input combination of deflections can be determined.

After completion of the analysis of loads and motions, a specification for an elastomeric bearing was prepared and the detail design, manufacture, and extensive bench tests of the bearings were subcontracted. Work was started in 1965 for bearings suitable for a new rotor blade that generated a centrifugal force of 67,000 pounds.

Upon completion of tests, the condition of the bearings was such that only minor modification to the design will be required to withstand the loads of the CH-47 advanced-geometry blade.

A prototype of a full-scale elastomeric-bearing hub assembly has been designed and built for use with advanced-geometry composite blades for a growth version of the CH-47 helicopter. The design and development problems are presented in this paper. The hub assembly was scheduled for whirl tower testing in the spring of 1968.

INTRODUCTION

Articulated rotor hubs traditionally consist of a complex structure with many elements in the load path from the rotor shaft to the rotor blades.

Conventional articulated rotors have three separate hinges per blade: one for pitch, one for lag, and one for flap. Each hinge usually has anti-friction bearings, a source of lubricants, and static and dynamic seals. Conventional hinge construction requires "clean" facilities for assembly, overhaul, and inspection. Field maintenance is difficult, and each hinge is susceptible to the effects of an adverse environment. Figure 1 shows schematically three basic rotor hinge arrangements, two of which are used at Vertol. In the first arrangement (Model 107-II), the pitch change occurs outboard of both flap and lag hinges, and the pitch arm leads and lags with the blade. The hinge pin does not tilt with pitch change.

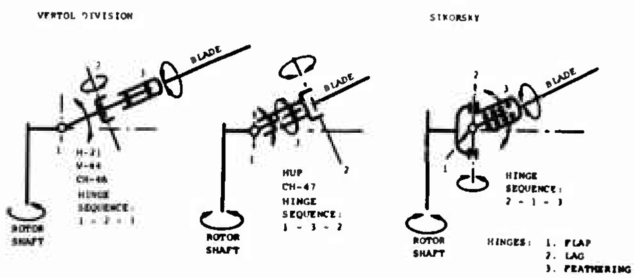

Figure 1. Typical Articulated Rotor Systems.

In the second arrangement (Chinook), the pitch change occurs between the flap and lag hinges, and the pitch arm does not lead and lag with the blade. The lag hinge pin tilts with pitch change.

The third system shown in Figure 1 is used in current Sikorsky helicopters. It also has the three hinges with the lag hinge inboard, the flap hinge next, with the pitch axis outboard. The pitch arm leads and lags with the blade.

The elastomeric-bearing rotor hub shown schematically in Figure 2 uses a single spherical bearing per blade to provide blade motion in the flap, lag, and pitch directions. The elastomeric bearing consists of concentric spherical laminations of an elastomer and metal bonded together in thin alternate layers. The bearing is mounted so that the centrifugal force exerted by the blade results in compression of the elastomer, while pitch, lag, and flap motions result in shear deflection of the elastomer. The layers of elastomer are thin enough to withstand high normal pressures, and the ability of each layer to deflect in shear (i.e., pitch, flap, and lag) is practically independent of normal pressure. Thus, the assembly can provide the required blade motion and still withstand high centrifugal loads.

In order to maintain the fixed geometrical location of the center of the elastomeric bearing regardless of alternating vertical and horizontal shear forces, a self-lubricating ("dry") stabilizing "coincident" or "shear" bearing is provided in such a manner that it resists the relatively light shear loads (normal to blade axis) only, while the high axial load (centrifugal force) is taken by the elastomeric bearing, which in turn does not feel any shear loads.

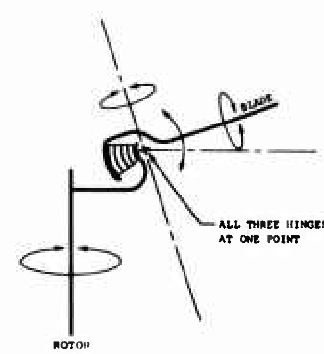

Figure 2. Schematic Drawing of Elastomeric-Bearing Rotor Hub.

Use of the elastomeric bearing results in a universal hinge that gives all three degrees of freedom while eliminating lubrication, seals, and relative motion between highly loaded contacting surfaces susceptible to dirt, water, and other contaminants. All motion occurs by deformation of the elastomer, so the assembly, reliability, inspection, and maintenance requirements are greatly simplified. The inherent mechanical simplicity of the elastomeric-bearing rotor hub may be seen in the axonometric projection of the three-bladed horizontal hook design known as the "Triskelion" hub (Figure 3). The simplicity of this hub concept is best

demonstrated by the comparison with a CH-47 type hub in Figure 4. Figure 5 illustrates the components in the rotor hub required to transmit the blade loads and motions. There are 408 such parts in the CH-47 hub arranged on three individual axes as compared to 48 parts performing the same work on the elastomeric-bearing hub. In addition, the CH-47 type hub has 233 more parts per hub to provide the necessary lubrication (see Figure 6) compared to none in the elastomeric-bearing hub design. Thus, 7 oil reservoirs, 15 sight glasses, 23 drain and fill plugs, 83 O-rings, and 24 oscillating seals plus associated retention parts are eliminated. The total number of components of the elastomeric-bearing rotor hub is reduced to 26 percent of that of the CH-47 hub.

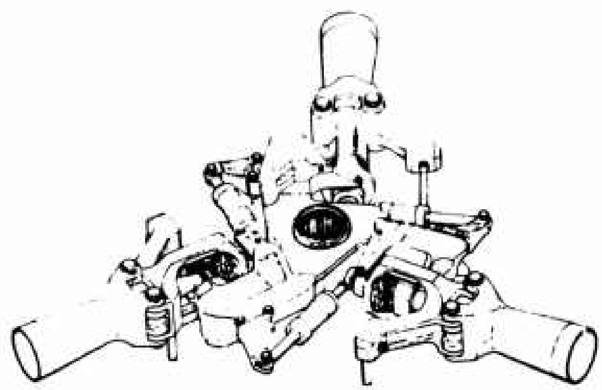

Figure 3. Three-Bladed Horizontal Hook Design (Triskelion) Elastomeric-Bearing Rotor Hub.

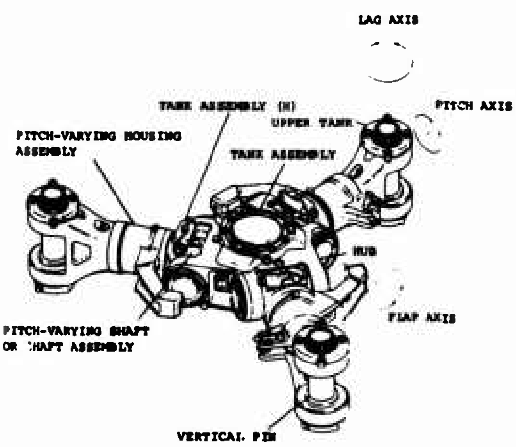

Figure 4. CH-47 Rotor Hub.

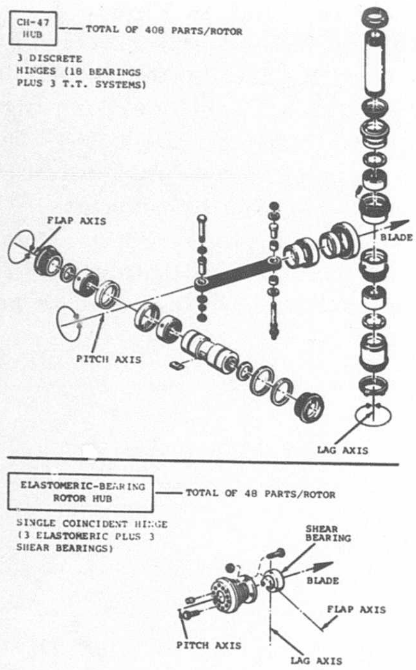

Figure 5. Comparison of Articulation Components.

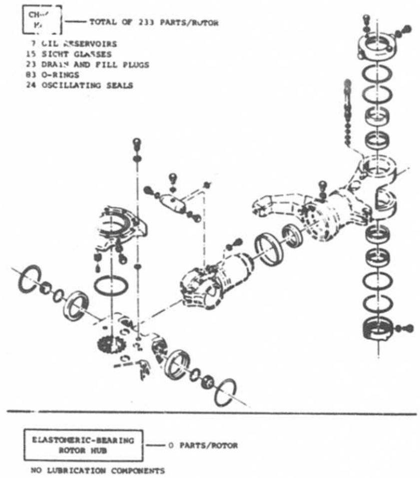

Figure 6. Comparison of Lubrication Components.

The technical discussion which follows is oriented towards use of a two-pin blade socket having attractive manual blade fold features. All comparisons made between Vertol's conventional and the elastomeric-bearing rotor hubs assume the use of the two-pin blade socket with the elastomeric-bearing rotor hub. However, to reduce development cost, facilitate blade replacement, and allow interchangeable use of advanced-geometry glass-fiber-reinforced plastic rotor blades and/or present droop-snoot blades without alteration, the standard single-pin blade socket is used in the prototype hub as shown in Figure 7. The use of standard blade sockets does not compromise the effectiveness of the elastomeric-bearing rotor hub.

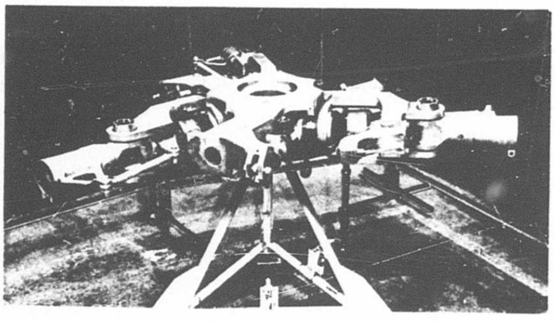

Figure 7. Prototype of Elastomeric-Bearing Rotor Hub.

IMPROVEMENT IN ROTOR TECHNOLOGY

The elastomeric-bearing rotor hub represents a new concept in rotor hub design. The development of an elastomeric-bearing rotor hub, including whirl tests and flight tests on a Chinook helicopter, combined with the new advanced-geometry blade, would contribute to the advancement of articulated rotor system technology.

Basically, the advanced-geometry glass-fiber-reinforced plastic blade offers improved rotor efficiency, which gives:

1. Improved range
2. Improved speed
3. Improved thrust

These results are achieved by better lift/drag ratios at high Mach numbers and advance ratios. Thus the advantages of the advanced-geometry blade are realized primarily at high flight speeds and gross weights. The aspects of advanced-geometry blades are fully discussed in reports associated with References 10 and 11; therefore, no further discussion of the blades is offered here. To the advantages of the advanced-geometry blade, the elastomeric-bearing hub adds:

1. Simplicity and thus reliability
2. Reduced maintenance
3. Increased control sensitivity
4. Reduction in vibratory shaking forces
5. Reduced cost
6. Reduced weight

Discussed below are the major advancements in rotor design represented by the elastomeric-bearing rotor hub.

AERODYNAMIC DRAG

Greater consideration must be given to aerodynamic drag in modern helicopters if higher flight speeds are to be achieved. In tandem-rotor

helicopters with unfaired hubs, the average rotor hub drag represents approximately 15 to 20 percent of total helicopter drag. Although there is no significant improvement in unfaired hub drag, the eventual potential of the elastomeric-bearing rotor hub lies in the use of a rotor hub fairing resulting in only 57 percent of the drag of the CH-47B hub.

Past studies of hub fairings for conventional articulated hubs showed that the three separate hinges of each blade required a fairing of such size that the increased cross section area cancelled out the reduction in drag coefficient, and the net drag was unchanged. The elastomeric-bearing hub lends itself to a much more practical fairing.

Wind tunnel tests of a fairing configuration show a drag reduction of over 40 percent of the drag of the unfaired CH-47 rotor hub. This makes the elastomeric-bearing hub the logical choice, aerodynamically, for a rotor system incorporating the advanced-geometry blade. However, the detail design, fabrication, and test of a rotor hub fairing are not included in the scope of the present developmental work.

CONTROL SENSITIVITY

Rolling Moment

Cyclic control response is reflected in a rolling moment on the fuselage for a given tilt of the rotor. Two factors influence this moment: the tilt of the thrust vector and the hub moment due to the hinge offset. The hinge offset of the elastomeric-bearing rotor hub is greater than that of the CH-47 rotor hub and increases the total control moment, in the roll sense, by approximately 40 percent for normal flight conditions. The increased moment due to hinge offset, shown graphically in Figure 8, is more than doubled.

The increased control moment will make the helicopter more responsive in roll, or a tradeoff can be made, keeping the helicopter response unchanged and reducing the lateral cyclic travel. The latter will reduce blade motions, reducing frequency and severity of droop stop contact.

Additional benefits are gained from a control standpoint when the elastomeric-bearing hub is used on a helicopter operating with an unloaded rotor. Rotor unloading will occur in the transition between powered flight and autorotation. In this period of reduced thrust, the principal part of roll control moment comes from the hinge offset, and so the elastomeric-bearing hub will give double the control response. The lift unloaded rotor in a high-speed compound helicopter will also

benefit from the increased rotor control moment and could eliminate the need for ailerons on these vehicles.

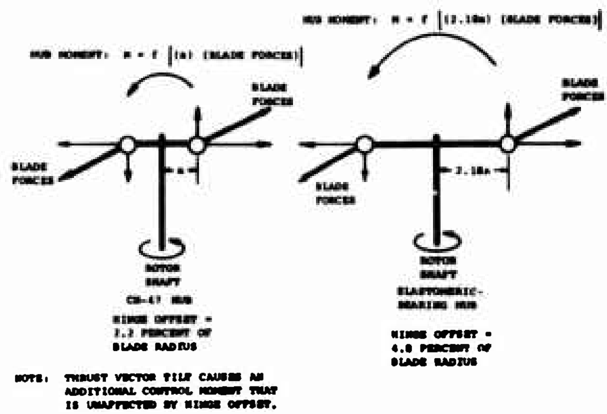

Figure 8. Effect of Hinge Offset on Control Moment for Elastomeric-Bearing Rotor Hub.

Roll Damping

Basic aircraft roll damping comes entirely from the rotor system. As with the roll control moment, it consists of contributions from thrust vector tilt and hinge offset (hub moments). Under normal flight conditions, the increased hinge offset of the elastomeric-bearing hub would increase the roll damping by about 50 percent. The damping capability of the elastomeric-bearing hub could be used to minimize dependence on roll stability augmentation systems.

The thrust vector tilt contribution to roll damping exhibits a marked reduction at high speed or high rate-of-climb conditions. Therefore, the increased hub moment contribution associated with the elastomeric-bearing hub would result in a more constant damping level under all conditions.

VIBRATION

Analysis shows that use of the elastomeric-bearing hub should result in a substantial reduction in three-per-rev vertical shaking forces. The three-per-rev vertical vibration is the most pronounced type of vibration mode in the CH-46 and CH-47 helicopters, and it comes mostly from root-end

flapping mass. In the conventional articulated hub, the flapping mass and effective flapping mass are identical. For the elastomeric-bearing hub, however, the flapping mass outboard of the hinge and up to 13 percent radius is 90 pounds less than that for a standard CH-47 hub. This means that the generated three-per-rev inertia shears are proportionately lower. In addition, the elastomeric-bearing hub has mass inboard of the flapping axis; i.e., the elastomeric bearing, the pitch housing, the damper attachment structure, and the lag damper. These masses on the inboard side seesaw with respect to the blade and actually create opposing inertia shears to further assist in reducing vibratory three-per-rev shears.

Figure 9 compares the effective flapping mass for the elastomeric-bearing hub with that for the conventional CH-47 articulated hub.

The calculated three-per-rev vertical shaking forces for the elastomeric-bearing and conventional CH-47 articulated hubs, both with advanced-geometry blades, are compared on a percentage basis in Figure 10. The baseline chosen for the 100 percent value is the 155-knot maximum speed vibration level. Figure 10 indicates that 10-15 knots higher forward speeds are obtainable with the elastomeric-bearing hub without increase in the vibration level.

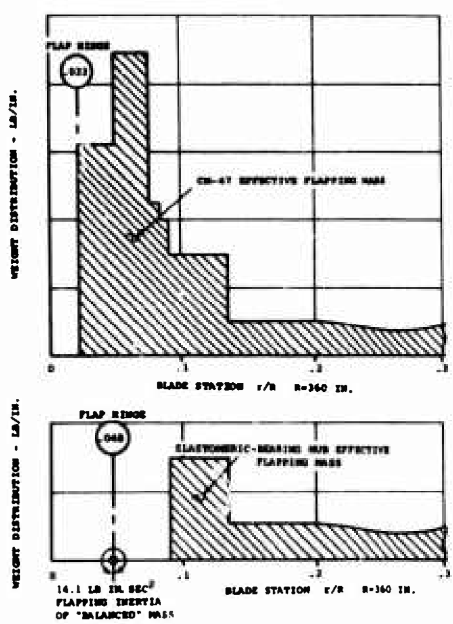

Figure 9. Comparison of Effective Mass Associated With Generation of Vertical Hub Shear Forces.

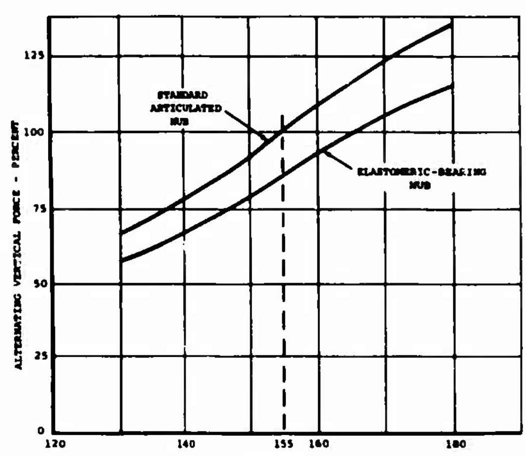

Figure 10. Vertical Shaking Force.

Blade-Hinge Motion Characteristics

The elastomeric-bearing hub has several specific features that differ from the conventional articulated hub. The following qualities of the elastomeric-bearing hub have been analyzed for possible effects on helicopter operation.

HINGE RESTRAINT

Effect on Loads

The antifriction bearings of conventional articulated rotor hubs give virtually no restraining moment to oppose blade hinge motion. However, the tension-torsion strap used in place of thrust bearings in the CH-47 helicopter to carry the blade centrifugal loads through the pitch hinge does act as a torsional spring with an approximate rate of 120 inch-pounds per degree. The elastomeric bearing has a spring rate of 165 inch-pounds per degree about the pitch axis, which is slightly higher than the tension-torsion strap, and, in addition, develops a moment proportional to blade displacement about the flap and lag axes. The magnitudes of these moments about each hinge axis for blade motions typical of the high-speed flight condition of the CH-47 are tabulated in Table I. The moment caused by the elastomeric bearing about the pitch axis (332 ± 995 inch-pounds) is less than 3 percent of the predicted blade pitching moment for the high-speed flight condition.

TABLE I. MAGNITUDE OF BLADE MOMENT ABOUT EACH HINGE AXIS

Blade Motion		Spring Moment of the Elastomeric Bearing (in. -lb)
Pitch	2° ± 6°	332 ± 995
Flap	0° ± 7°	0 ± 3670
Lag	3.8° ± 1.7°	2000 ± 890

Vibratory blade flap moments, including the influence of the restraining moment of the elastomeric bearing of ± 3670 inch-pounds at station .08R (just outboard of the flap axis), are shown in Figure 11. These moments are up to 30 percent lower than those of a corresponding station for the conventional articulated CH-47 hub. The influence of hinge restraining moments diminishes at more outboard blade stations and therefore can be considered negligible.

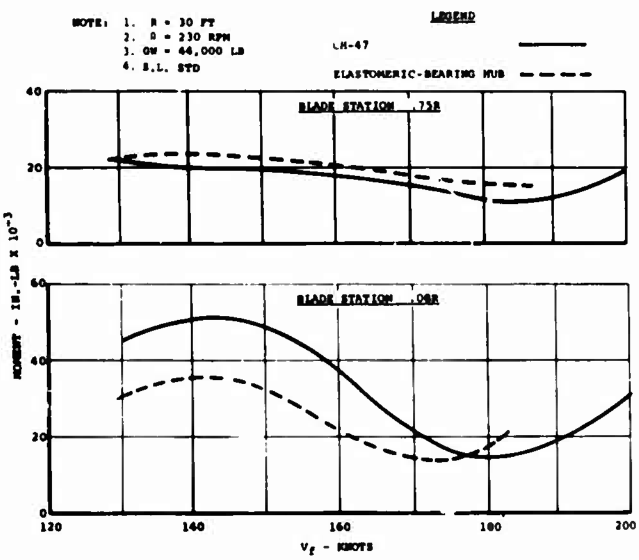

Figure 11. Vibratory Flap Moments of Advanced-Geometry Plastic Blade at Various Stations.

The moment about the lag hinge axis of the elastomeric bearing (2000 ± 890 inch-pounds) is also low in comparison to other moment sources and will have negligible effect on the strength of components.

Aeroelastic Vibration Response

A theoretical study employing the Leone-Myklestad method was made to determine the free and forced aeroelastic vibration response of an advanced-geometry fiber-glass rotor blade attached to an elastomeric-bearing rotor hub. The analysis included the effect of angular spring restraints of equal magnitudes about the mutual orthogonal flap and lag axes which are coincident at the attachment point.

The free vibration analysis yielded blade natural frequencies in the flapwise and chordwise directions which are quite similar to those associated with rotor blades on a fully articulated rotor hub because of the relatively small magnitudes of the root angular spring constants. Similarly, the

results of the forced vibration analysis for high-speed level flight showed the blade root flap bending, chord bending, and torsional moment to be virtually the same as those of blades on an equivalent fully articulated rotor hub. Again, this was because of the relatively small magnitude of the root angular spring constants, where, for example, the vibratory flap moment about the elastomeric hinge is less than 2 percent of the maximum blade moment. Figure 12 shows the variation of natural frequencies for various hub-blade combinations.

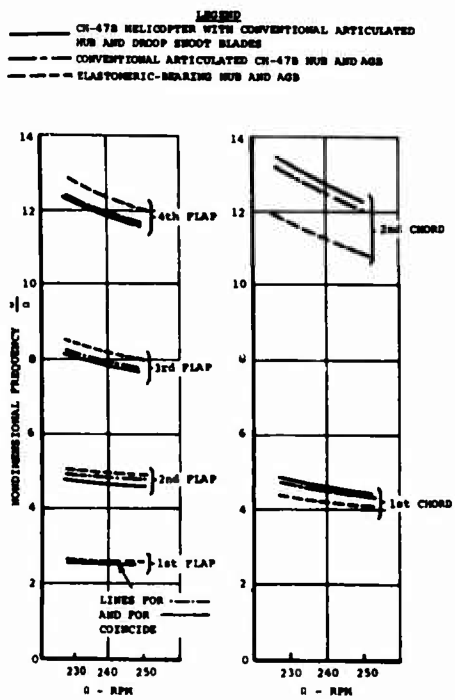

Figure 12. Variation of Uncoupled Natural Frequency With RPM for Various Hub-Blade Combinations.

The general conclusions of the theoretical study are that the elastomeric-bearing rotor hub will yield a rotor blade aeroelastic response which is virtually the same as that produced by an equivalent conventional articulated rotor hub.

LAG DAMPER MOTION

Use of the single spherical bearing to accommodate blade flap, lag, and pitch motion makes it mechanically complex to mount the lag damper in such a way that it is affected only by blade lag motion and not by pitch and flap. Kinematic analyses were made of various lag damper attachment

locations, and a satisfactory arrangement was found to be inboard of the elastomeric bearing and on the pitch axis of the blade. This location made lag damper operation independent of blade pitch.

Blade flap motion does affect lag damper motion very slightly, but analysis of the damper forces resulting from a blade path (flap and lag) typical of high-speed flight showed a slight distortion of the damper force curve (Figure 13) similar to the distortion associated with the conventional articulated hub employing linear dampers subjected to the cyclic variations in effective mounting radius. The damper attachment point on the pitch axis which has been chosen for the elastomeric-bearing hub is shown in Figure 14. For ground instability considerations, the damper action is no different from that of the conventional articulated hub, as this phenomenon is essentially an in-plane blade motion.

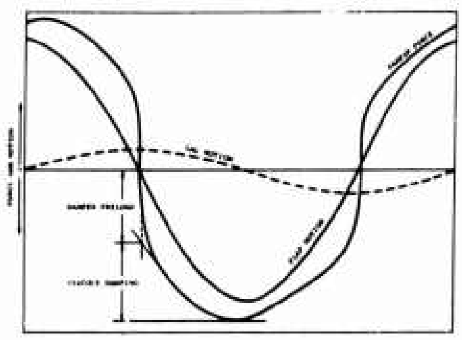

Figure 13. Damper Force During One Revolution of Blade.

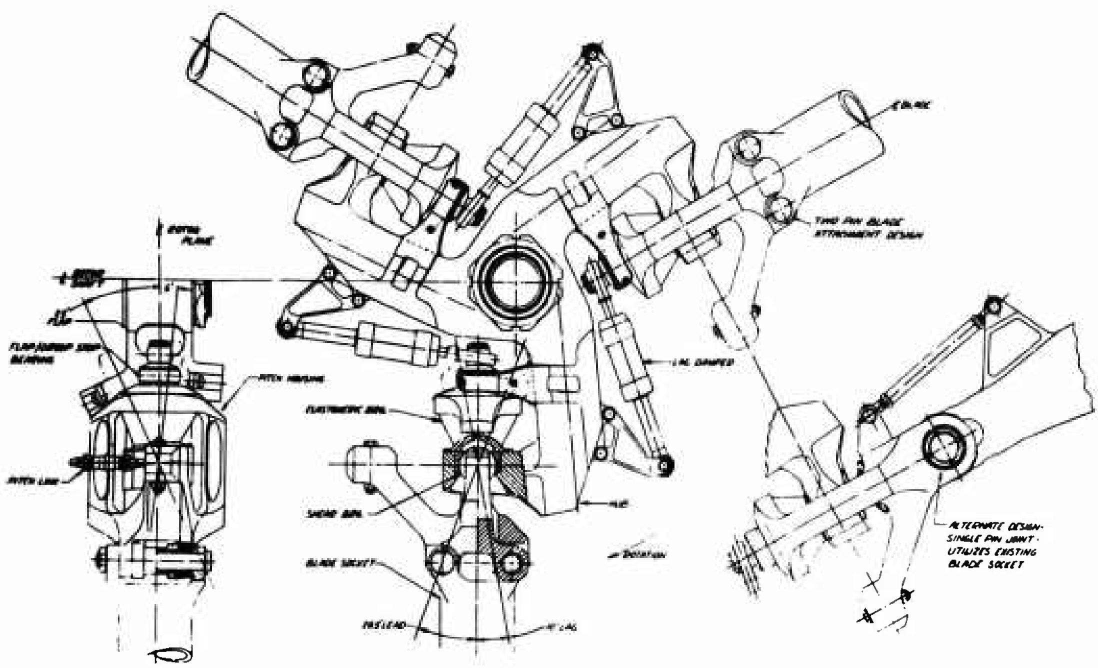

Figure 14. General Configuration of Elastomeric-Bearing Rotor Hub.

BLADE CLEARANCES

The elastomeric-bearing hub gives improved rotor blade-to-ground clearances and blade-to-fuselage clearances over the CH-47 articulated hub for stationary blades (ground flapping) and at low rotative speeds.

The lag hinge axis of the CH-47 tilts with pitch change and is normal to the rotor plane at 10.5 degrees blade pitch, the approximate collective pitch setting required for hover. The elastomeric-bearing hub does not have a tilting lag axis characteristic, so an additional 2 degrees of droop can be permitted and still retain the blade-to-fuselage and blade-to-ground clearances of the conventional articulated CH-47 rotor. A further improvement in clearances is obtained because the static droop of the advanced-geometry blade is only two-thirds that of the conventional rotor blade.

Accordingly, the droop stop setting for the elastomeric-bearing hub with advanced-geometry blades can be lowered by 2.5 degrees and still retain better blade/fuselage clearances than the CH-47A, B, or C helicopters.

The improved clearances as applied to the present CH-47 helicopter would be of benefit in two ways. First, increased blade downward flapping motion of the forward rotor will be possible without droop stop contact; second, the centrifugal droop and flap restrainers on the aft rotor can be eliminated, resulting in a saving of weight and cost as well as simplification of design.

SYSTEM EFFECTIVENESS

The elastomeric-bearing rotor hub is a considerably more efficient system than the conventional articulated rotor hub. Overall reliability and mean time between failure for the elastomeric-bearing hub appear to be 50 percent greater than those for the CH-47 hub. The elastomeric-bearing hub design has inherent safety characteristics which are not present in contemporary conventional articulated rotor hubs. Maintenance requirements, and thus maintenance time, are greatly reduced. The amount of special ground-support equipment necessary for maintenance of the elastomeric-bearing hub is much less than that presently needed for the CH-47 hub. A further indication of the design effectiveness of the elastomeric-bearing hub is its potentially reduced acquisition cost.

RELIABILITY

Elimination of the hinge bearing lubrication requirements results in improved reliability for the elastomeric-bearing hub. Reliability statistics

on the rotor heads of the CH-46 and CH-47 helicopters have been carefully monitored in over 8000 hours of flight operation. These studies show that the lubrication system accounts for 40 percent of the rotor head primary malfunction rate on the CH-47 and 70 percent on the CH-46.

To date, endurance of the elastomeric bearing has proven to be very satisfactory. Mechanical damage occurred to one bearing when excessive motion was inadvertently introduced prior to endurance bench testing. However, the bearing was functionally satisfactory and continued to be so throughout the 1200-hour endurance testing that followed. An observable progressive deterioration did occur that would have been inspectable on the rotor hub, and it is probable that observation of this condition would have led to replacement of the bearing. Therefore, based on the endurance testing to date, the mean time before failure has been conservatively set at 2400 hours.

The reliability of the elastomeric-bearing hub is such that, based on the above experience and the assumption that all other component failure rates are unchanged, the mean time between failure is 50 percent greater than that for the CH-47 hub. It is probable that the reduced number of components in the primary load path will improve the reliability still further, but this cannot presently be quantitatively established.

The design attributes relative to safety are:

1. Reduction in the series load path and corresponding reduction in the number of prime structural components

2. Simplification as evidenced by the significant reduction in total parts

3. Elimination of the lubrication interface

4. Ease of manufacture and fabrication

5. Ease of maintainability and service

6. The decrease in possible error or negligence during manufacture and maintenance which is associated with the above five items

Finally, the most significant attribute is the inherent fail-safe characteristic of the bearing. Failure of spherical elastomeric bearings is of the slow progression type, which will permit the bearing to maintain its

proper function, and is such that the failure could be detected within reasonable inspection periods. Even a complete failure of the elastomer does not result in blade separation from the rotor hub.

In tests, the bearings have shown the ability to function properly with the design loads and motions even though noticeable degradation of the external areas of elastomer has occurred. Such damage progressed at a slow rate similar to visible abrasion, although the bearing was structurally satisfactory at the conclusion of the 1200-hour endurance test.

A second bearing experienced a similar abrasion when deliberately cycled at 40 percent beyond its design load for 320 hours of testing. Early traces of this abrasion were visible after 24 hours of testing; again, the progression of failure was such as to be easily detected over reasonable inspection intervals.

A bearing qualification program continued beyond the present scope would give:

1. Assurance that potential manufacturing defects or inconsistencies which might be undetected by normal quality control procedures or nondestructive testing/examination are not a potential hazard

2. Assurance that environmental requirements are met

MAINTAINABILITY

A maintainability analysis indicates that, compared with the conventional CH-47 rotor hub, the elastomeric-bearing rotor hub offers the following major advantages:

1. Scheduled inspection takes less time

2. Overhaul hours are reduced 75 percent

3. Environment resistance is improved

4. Spares requirements are reduced

5. Overall man-minutes per flight-hour for maintenance are reduced

MAINTENANCE MAN-MINUTES PER FLIGHT-HOUR

The elastomeric-bearing hub has approximately 74 percent less parts than the present production head. This reduction in parts is accomplished by the elimination of the horizontal and vertical pin assemblies, the tension-torsion strap and pin assemblies, the pitch-varying housing, and the rotary-wing pitch shafts; associated attendant hardware and the necessity for lubrication are also eliminated.

With fewer parts and less complicated installation, the scheduled inspections will require less time to accomplish. It is estimated that the maintenance man-minutes per flight-hour required for the CH-47 hub will be reduced to one-fourth of its present value for the elastomeric-bearing hub.

GROUND-SUPPORT EQUIPMENT

The design for the elastomeric-bearing rotor hub will reduce the recommended quantity of special ground-support equipment (GSE) from 32 items to approximately 10 for all levels of maintenance. Of the 10, 7 will be similar to existing GSE, 1 will require no change, and 2 will be of a new design.

SPECIAL TOOLS

The majority of special tools are required for bearing and oil seal removal and installation and would be eliminated.

The proposed elastomeric-bearing rotor hub would require the following maintenance items used on the present CH-47 rotor hub configuration to be either modified or replaced:

- Workstand
- Adapter
- Adapter set
- Lift device
- Balancing adapter
- Text fixture

The maintenance crane used on the present rotor hub configuration is suitable for use on the elastomeric-bearing rotor hub. The following new maintenance items may be required for the elastomeric-bearing rotor hub:

Wrench, bearing retainer nut
Workstand, rotor balancing

ACQUISITION COST

Comparative cost data have been compiled for quantity production of the steel version of the elastomeric-bearing rotor hub. The data show cost savings averaging nearly 10 percent as compared with the CH-47A articulated rotor hub for quantities of 300 ship sets.

STRUCTURAL CONSIDERATIONS

The structural substantiation of the design in Figure 14 was made in accordance with MIL-S-8698, Structural Design Requirements, Helicopter (Reference 6).

The elastomeric-bearing hub is designed to absorb 7500 horsepower in a 60-40 percent distribution of power applied to the two rotors of a tandem helicopter of 44,000 pounds gross weight. Growth to 9000 horsepower is provided for; that is, provisions can be made within the basic design envelope to absorb two rotor powers of 9000 horsepower with no change in basic geometry and only local strengthening. Three advanced-geometry blades with a centrifugal force of 93,000 pounds at the normal rotor speed of 230 rpm are to be used. The predicted spring rates for the elastomeric bearings are: flap or lead-lag = 30,000 inch-pounds per radian; pitch = 9500 inch-pounds per radian.

With careful attention given in the initial detail-design-drawings stage to details of fretted areas and at discontinuities, a design that is adquate for steel can be made satisfactory for titanium with only minor changes.

WEIGHT

The elastomeric-bearing rotor hub assembly with titanium structural members weighs only 87 percent of the CH-47C rotor hub assembly, and the complete rotor system (advanced-geometry composite blade with the elastomeric-bearing rotor hub) saves about 85 pounds per rotor over the present CH-47C rotor system.

It should be noted, however, that the present CH-47C rotor hub, although usable for flight demonstrations of the advanced-geometry blade, requires

strengthening in order to sustain continuously the loads of the growth CH-47 as defined above. Substantial weight savings can be expected for the elastomeric-bearing hub using titanium. Hub weights are 658 pounds for the elastomeric-bearing hub versus 813 pounds for the titanium version of the CH-47 type hub.

The elastomeric-bearing hub has considerable potential for additional weight reduction as well as an increased load potential exceeding the capability of the CH-47 rotor head. The production detail design phase of hub development should produce further weight reduction. However, some weight increases caused by increased hub moments of the rotor shafts and rotating controls can be expected.

At present, it is not possible to refine the design to obtain any weight-load optimization. The prototype of the elastomeric-bearing hub is not representative for this purpose, as it contains several simplifications and departures from the future production version. Therefore, the weight comparison data for the elastomeric-bearing rotor hub design have been obtained from the structural substantiation analysis. The weight comparison data for the CH-47C hub, on the other hand, represent refinements resulting from many man-years of design effort as well as the knowledge gained from a complete bench endurance test program, whirl test program, and flight strain surveys. Thus, the weight data of the elastomeric-bearing hub are considered to be conservative.

DESIGN OF THE ELASTOMERIC BEARING

The elastomeric bearing performs four functions:

1. Carries the full centrifugal force
2. Accommodates pitch motion
3. Accommodates flap motion
4. Accommodates lead-lag motion

Performance of all four of the above functions simultaneously by a single bearing requires a spherical configuration for the bearing. Figure 15 shows such a spherical elastomeric bearing. The bearing is designed for a centrifugal force of 67,000 pounds and is one of eight bearings that have been used for testing purposes. The normal pressures and elastomer strains in this bearing are typical of those that will be applied to a bearing

on the CH-47 hub with advanced-geometry composite blades; therefore, test data relating to spring rates are representative of the order of magnitude anticipated for the CH-47 elastomeric bearing.

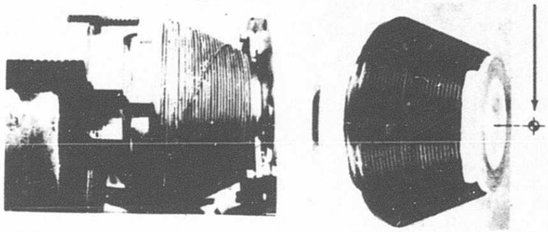

Figure 15. Elastomeric Bearing.

The bearing in Figure 15 is constructed of alternate spherically shaped layers of steel and rubber; however, titanium is considered for production versions. Pitch, flap, and lead-lag motions force relative movement of the metal shims, which are restrained by shearing forces provided by the elastomer layers to which the steel shims are bonded. The accommodation of motion by shear forces in elastomer elements rather than by antifriction bearings is extremely attractive for four reasons:

1. Reduction in number of parts
2. Reduced maintenance
3. Absence of rubbing, wearing, or rolling of rolling elements
4. Elimination of contamination by such environmental matter as dirt, dust, or water

The centrifugal force is countered by compression loads on the bearing laminates. The bearing has a very high spring rate in compression, higher than predicted, resulting in a very low deflection under compressive loads, which is of benefit to the design. Pitch spring rate and the flap spring rate are both low and in good agreement with predicted values.

Figure 15 shows the bearing with a torsional pitch wind-up of 155 degrees. The almost perfectly helical shape assumed by the seam on the bearing shows that it is possible to design the bearing such that the strain is evenly distributed among the laminates.

Since the elastomeric bearing is required to carry by compression the centrifugal force generated by the rotating rotor blade, its column stability must be assured over the operational load and motion spectrum. The stability aspects for the elastomeric-bearing hub have been successfully investigated experimentally.

Compressive loads up to 115,000 pounds were applied to the bearing, and it was then deflected to 10 degrees in the flap or lag plane. No instability was noted. A bearing under centrifugal force of 67,000 pounds and 15 degrees flap deflection is shown in Figure 16.

Although the elastomeric bearing is normally required to withstand the compressive loads, very small tensile loads may occasionally occur as the result of ground handling and flapping of the blade. Accordingly, a test was run to determine the bearing's tension capability. The ultimate tensile strength of the bearing was found to be equivalent to approximately twenty times the blade weight, thus providing ample margin.

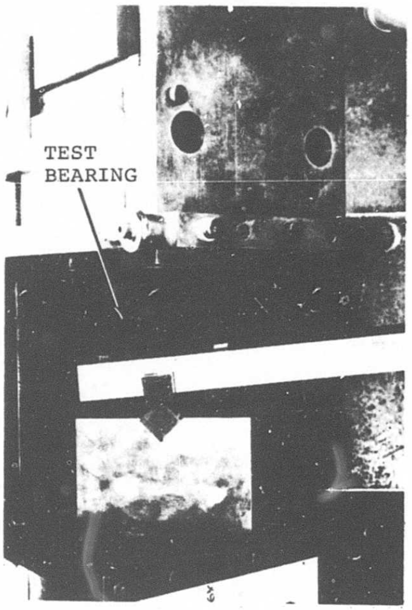

Figure 16. Single Bearing Stability in Compression on Knife Edge.

GEOMETRY, SHEAR STRAIN, AND SIZE

The primary design factor affecting a laminated elastomeric component is the magnitude of shear strain to which the rubber layers are subjected. In addition to the material properties, there are two factors which affect shear strain in a laminated bearing: (1) shape factor and (2) geometry of the component. The shape factor governs how much strain is contributed by laminate bulging under applied compressive loads. The effect of bulging on shear strain is difficult to determine and is considered small with respect to shear deformation due to motion input. The geometry of the bearing affects the amount of strain per layer for a given angular deflection of the bearing.

A method for theoretical analysis of spherical elastomeric bearings and a digital computer program have been set up to determine elastomeric

strains for any bearing motion. To complement these, a more extensive analysis, one which permits vector resolution of strains caused by blade pitch, flap, and lag motions, has been used. This latter analysis has also been transformed into a computer program so that mean and alternating strains at any location in an elastomeric bearing can be determined rapidly.

Figure 17 shows index shear strain at four points on the periphery of a bearing, 90 degrees apart. The combined strain of each of these points resulting from simultaneous application of pitch, lag, and flap motion, under conservative phasing assumptions, and assuming simple harmonic motion during one rotor revolution, is plotted in a polar diagram.

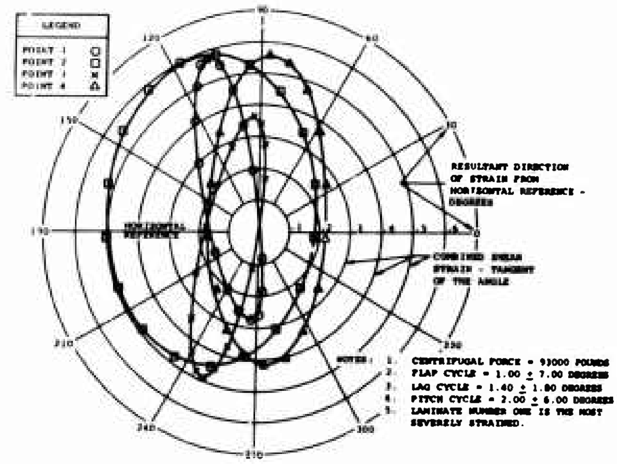

Figure 17. Index Shear Strain at Four Points on the Periphery of an Elastomeric Bearing.

Further analytical investigation to determine the magnitude of tensile stresses in the bearing caused by deflection in the flap or lag direction is planned.

ELASTOMER

In selecting materials for any given application, advantages and limitations of candidate materials must be considered. The ideal material for a given application is the one whose advantages best meet the design requirements and whose limitations are not realized in the application or can be minimized by proper design.

In selecting an elastomer for the bearing, it was decided that natural rubber had the proper advantages for the intended function and that its

limitations did not present serious design problems. The following characteristics were considered in the selection of natural rubber for the elastomeric bearing:

1. Advantages

   a. High strength
   b. Excellent fatigue characteristics
   c. Accumulated experience and knowledge gained through years of use in the aircraft industry

2. Limitations

   a. Restricted operating temperature range
   b. Susceptibility to light, ozone, and oil contamination
   c. Aging

Since the elastomeric bearing is subject to high loads in a fatigue environment, the advantages cited for natural rubber are ideal for the application. However, the limitations for this specific application should be examined.

Operating Temperature Range

By varying the compounding of natural rubber, the useful temperature range can be from -65°F to 180°F. The required operating temperature range of the hub assembly is -65°F to 160°F. Elevated temperatures of about 160°F are not unusual in the use of natural rubber compounds. There are many current engine suspension systems that operate continuously in environments in excess of this temperature. To add to the understanding of the effect of elevated temperature on fatigue strength, Figure 18 shows typical load deflection results before and after 500 hours of dynamic testing at 200°F. The curve, plotted after 500 hours of fatigue testing, shows that the spring rate of the part is still within production tolerances, indicating little change in expected performance. At the -65°F extremity of the operating temperature range, consideration must be given to the increase in stiffness

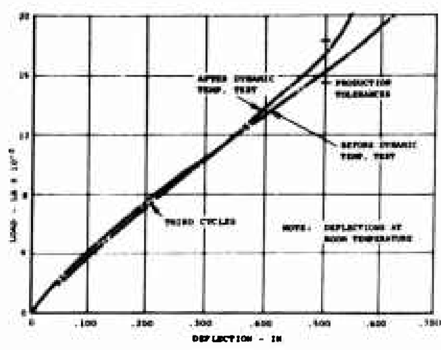

Figure 18. Typical Load Deflection Results Before and After 500 Hours of Dynamic Testing at 200°F.

of the rubber. (Brittleness is not a problem since the brittle point of natural rubber is about -80°F.)

Dynamic tests conducted on shear test samples of natural rubber compounds at -65°F show that while the stiffness increases on the order of 22 times the stiffness existing at room temperature, the part is capable of accommodating motion without fracturing. Referring to Figure 19, it will be seen that flexing of the test sample at speeds and amplitudes typical of a rotor blade environment reduced the stiffness ratio from an initial 22 times the room temperature value to approximately 8.5 in five cycles, or 1 second of flexing, and indeed the part approaches the room temperature value after only 10 minutes of operation. The values for a test bearing run at a slightly lower frequency are initially somewhat higher, but after 300 cycles approach that of the shear test sample. On the other hand, at low frequencies, although the control system in the helicopter will be initially stiff after a -65°F soak, control system check-out prior to flight and cyclic flexing prior to lift-off can reduce the stiffness by about 25 percent. The control system check-out prior to takeoff after a -65°F soak will not result in damage to control components, such as the pitch link, since the hydraulic system limits applied forces.

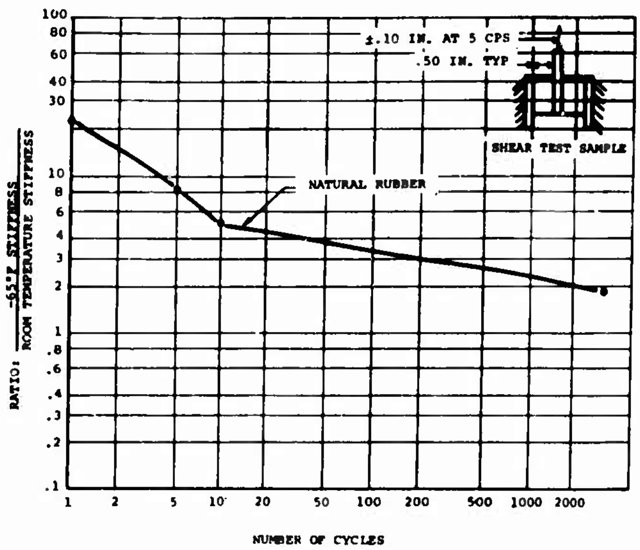

Figure 19. Stiffness Ratio of Natural Rubber.

The effect of start-up at -65°F on the fatigue life of the control components has also been investigated and found to be negligible. If it is assumed that a growth Chinook helicopter with elastomeric-bearing hubs is started at -65°F with full-down collective pitch (4 degrees) and -6 degrees longitudinal cyclic trim, the first cycle pitching moment due to deflection of

the elastomeric bearing will be 13,600 ± 21,900 inch-pounds, which results in a first cycle pitch link load of 1330 ± 2130 pounds for a 10.25-inch pitch arm radius.

The stiffness of the rubber decreases rapidly with increasing cycles, causing the pitch link loads to decrease as shown in Table II. After 100 cycles (less than 1 minute of cycling), the pitch link load has reduced to 205 ± 330 pounds, which is less than 8 percent of the pitch link fatigue design load.

TABLE II. RELATION OF STIFFNESS AND PITCH LINK FORCE TO NUMBER OF CYCLES

No. of Cycles	Dynamic Stiffness Relative to Room Temp.	Pitch Link Force (lb)
1	22 times	1330 ± 2130
5	8.2 times	495 ± 795
10	5 times	305 ± 485
100	3.4 times	205 ± 330
3500	1.8 times	110 ± 175

Light, Ozone, and Oil Contamination

One of the major advantages of the elastomeric-bearing rotor hub concept is the fact that no lubrication is required. The danger of oil contamination of the elastomeric bearing is, therefore, almost eliminated. The only fluid normally present in the rotor head area would be the hydraulic fluid in the lag damper. To protect against any possible contamination from this source, as well as surface exposure to light and ozone (if such a protection is at all required), a protective coating for the bearing is being considered. Various coatings have been tested in conjunction with the bearing test program.

Aging

Natural and some synthetic rubbers show changes in properties with the passage of time. The magnitude of this change with a natural rubber varies with the base stock and the compounding. Sensitivity to aging is

also increased by exposure to sunlight, ozone, oxygen, heat, rain, and similar environmental factors experienced in the normal course of manufacture and use. As is the case with many items in the rotor head, age control of the elastomeric bearing will have to be governed by provisions similar to those outlined in ANA Bulletin Number 438C (Reference 3). This will control storage conditions and limit accumulated age from the cure date of the bearing until the rotor head with this bearing is installed in the helicopter.

During service, the environment cannot be controlled as it can be during storage, and the tendency of the elastomer to age will be at its greatest. Figure 20 shows a typical load-deflection curve for a natural rubber elastomer mounting before and after 5 years' service exposed to extremes of temperature, weather, and atmospheric contaminants. The mounting was used for supporting a motor generator under a railroad car. This mounting was constructed of a rubber of similar composition to that used for the elastomeric bearing. The elastomer did not have a coating for environmental protection.

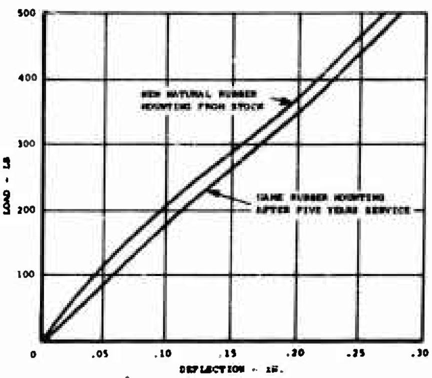

Figure 20. Aging of Rubber Components.

INTERNAL DAMPING

The internal damping of elastomers is an order of magnitude greater than that of metallic materials, and this property was examined to determine any possible influence it would have on rotor hinge application. Two factors were considered:

1. The additional damping introduced about the pitch, flap, and lag axes

2. The possibility of excessive temperature rise which might overheat the bearing

The following discussions reveal these factors to be of negligible importance.

Damping About the Pitch, Lag, and Flap Axes

Force-displacement records taken during tests of natural rubber elastomeric thrust bearings show the hysteresis loops with a phase angle between force and displacement of 6 degrees during room-temperature cycling and 19 degrees during testing at -65°F. A natural rubber elastomer was tested with normal pressures and cyclical strains in the same order of magnitude as projected for current applications. Data on other natural rubber elastomers show lower hysteresis, so the 6-degree value used in the succeeding analysis should be conservative. Damping of synthetic elastomers is also of the same order of magnitude.

Experimental evaluation of elastomeric bearing spring rates at room temperature for the test bearing gave the following data:

Pitch spring rate	114 in.-lb/deg
Lead-lag, flap spring rate	335 in.-lb/deg

Internal work computed for the high-speed level-flight blade motions is given in Table III for a 6-degree phasing between force and displacement, equivalent to normal temperature operation. These values of internal work are compared with typical energy levels on the blade for pitch, flap, and lag operation. As can be seen, the internal energy generated by hysteresis of the elastomeric bearing is small compared to the general level of energy in the blade. Damping about the lag and pitch axes is beneficial although small in magnitude.

TABLE III. INTERNAL WORK VALUES COMPARED WITH TYPICAL ENERGY LEVELS

Condition	Motion	Internal Work/Cycle (Elastomeric Bearing)	Reference Energy Level
Cyclic Pitch	± 6°	25 in.-lb	10,000 in.-lb*
Cyclic Flapping	± 7°	95 in.-lb	100,000 in.-lb**
Cyclic Lag	± 1.7°	5 in.-lb	2,000 in.-lb***

*Approximate energy dissipated by aerodynamic damping about the pitch axis per rotor revolution.

**Approximate kinetic energy of the blade equivalent to a one/rev flapping of ± 7 degrees at rotor speed.

***Approximate lag damper energy dissipation per rotor revolution equivalent to one/rev motion of ± 1.7 degrees at rotor speed.

Temperature Rise in the Bearing

During endurance testing of the bearings under a typical load and motion schedule, bearing temperature stabilized at approximately 30°F above ambient temperature with no cooling of any sort (see Figure 21). With the cooling environment of normal rotor head operation, the temperature

rise would be even less, and far from a value which would cause degradation of the bearing.

As discussed previously, the hysteresis in the elastomer is much greater at -65°F ambient than at normal temperature. Since the elastomeric-bearing spring rate is also increased at -65°F, this increased hysteresis is beneficial, as the bearing temperature rise will be increased and give the effect of operating at a higher ambient.

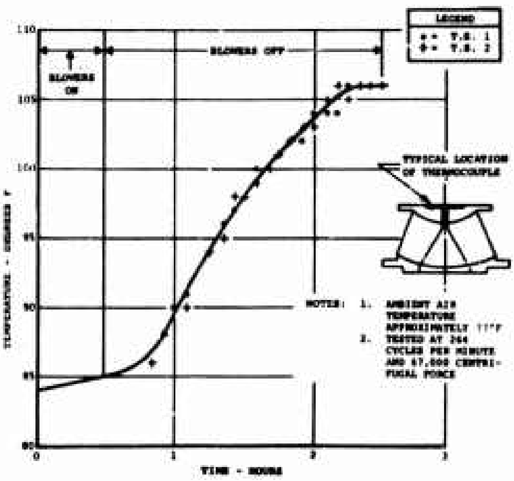

Figure 21. Temperature Stabilization During Testing.

QUALITY ASSURANCE

Present specifications define inspection requirements and testing for critical and semicritical dynamic parts and assemblies and now apply to the major structural components of the rotor head and blade; such specification requirements would apply directly to the elastomeric bearing. The bearing and its components would be categorized accordingly as follows:

1. Complete bearing - critical part
2. End plates - critical part
3. Shims - semicritical detail

All critical parts require destructive and nondestructive tests and test records starting with the raw material and continuing until the part is

completed. All critical parts would be serialized. A destructive evaluation would be made of a random sample from the first lot.

Semicritical parts would require nondestructive tests and test records from the raw material to the finished component.

Typical of the inspection and controls which the elastomeric bearing would receive in production as a critical part are the following manufacturing process and final product controls.

The specification for process control of the elastomeric bearing will use the same concepts of control as those used for rotor blade bonding. Process control standards now used for control of structural bonding processes and inspection for rotor blade construction follow the intent of MIL-A-9067 (Reference 2), and as applied to the elastomeric bearing will include:

1. Periodic check of the adhesive system

2. Control of manufacturing processes through destructive coupon tests for detection of process variables

3. Heat and pressure records of all vulcanizing and bonding operations

4. Random part samples to be tested to destruction (rate of sampling to be established)

5. Nondestructive examinations of every bearing as follows to insure part integrity:

    a. Visual and/or other suitable inspection will be made for voids (advanced methods like ultrasonic inspection will be considered).

    b. Compression load-deflection curves will be recorded, with spring rate tolerance to be ± 15 percent.

    c. Pitch (torsional) load-deflection curves will be recorded, with spring rate tolerance to be ± 15 percent.

    d. Proof load-deflection test will be made in pitch to a strain value equivalent to the maximum that the bearing could possibly experience in the application. While deflected, the bearing will be visually inspected for bond integrity.

The bearing will be deflected in the flap-lag direction also, and while maintaining this deflection, the bearing will be rotated so that the entire surface can be inspected for bond integrity. These tests will be run with no centrifugal force load applied.

In addition to the above, a log record of each bearing would be established.

ELASTOMERIC-BEARING TESTS

Since the elastomeric bearing is the only hinge connection between the blade and the hub structure, it was put through a rigorous test program in order to insure that it will perform its intended function and withstand the fatigue and environmental conditions present in a helicopter rotor system. The test program duplicated as closely as possible all of the conditions that the elastomeric bearing would experience in actual flight. The bearings tested have a design centrifugal force of 67,000 pounds. The test results will apply to the larger CH-47 growth elastomeric bearing, as the design, pressures, and strains on the elastomer are identical.

The test program included a total of eight test bearing samples (see Table IV). The eight bearings were subjected to fatigue testing according to the flight conditions specified in Table V, conditions 1-6. This fatigue schedule was run at room temperature, low temperature (-65°F), and elevated temperatures (125°F and 160°F). Conditions 7 and 8 represent maximum design excursions. The fatigue test machine is shown in Figure 22 with two test bearings in back-to-back arrangement. Figure 23 shows the schematics of the test machine.

TABLE IV. TEST PROCEDURE FOR THE ELASTOMERIC BEARINGS

Test Samples	Test Sequence			
1 and 2	400 Hours Room Temp. Fatigue Conditions 1-6	400 Hours Room Temp. Fatigue Conditions 1-6	200 Hours Room Temp. Fatigue Conditions 1-6	200 Hours Room Temp. Fatigue Conditions 1-6
3 and 4	200 Hours -65°F Conditions 1-6	200 Hours +125°F Conditions 1-6	200 Hours Room Temp. Fatigue Conditions 1-6	200 Hours Room Temp. Fatigue Conditions 1-6. Start-ups without CF at +160° and -65°F 100 times each.
5 and 6	Maximum Brg Excursion Room Temp. Conditions 7-8	800 Hours Room Temp. Fatigue Conditions 1-6	200 Hours Room Temp. Fatigue Conditions 1-6	
7 and 8	Spring Rate and Cycling Tests at -25°F	Spring Rate and Cycling Tests at -40°F	Fatigue at 39% Increased Centrifugal Force (Growth CH-47 load) Conditions 1-6	

TABLE V. ENDURANCE TEST CONDITIONS FOR ELASTOMERIC BEARINGS

Conditions	Description	Percent Time	Speed CPM	Centrifugal Force (lb)	Flap Angle (deg)	Lead-Lag Angle (deg)	Pitch Angle (deg)	Requirements of 100-Hour Block Test	No. of Cycles Required for 1000 Hours
1	High-Speed Level Flight	20	264	67,000	+1 ± 7	+1.4 ± 1.7	+2 ± 6.0	20 Hours	3.170×10^6
2	Maximum Power Climb	6	248	60,000	+1 ±4.5	+1 ± 1.0	+2 ± 4.0	6 Hours	0.892×10^6
3	Cruise	50	264	67,000	+1 ±6.3	-.13 ± 1.5	-1.5 ± 5.0	50 Hours	7.920×10^6
4	Transition	10	264	67,000	+1 ±4.5	-.87 ± 1.0	-3 ± 3	10 Hours	1.585×10^6
5	Hover (OGE)	10	264	67,000	+1 ±4.5	-1.0 ± 1.0	-1.5 ± 2.0	10 Hours	1.585×10^6
6	Autorotation	4	300	87,500	-1 ±4.5	-5.6 ± 1.5	-11 ± 3	4 Hours	0.720×10^6
7	Max in Flight*		slow	67,000	0 +20	0 -15	0 +45.5		100 Cycles
8	Rotor Braking*		slow	40,000	0 -10	0 +15	0 -10		10,000 Cycles

Conditions indicated by * are to be in phase and are considered as ultimate loads. For all other conditions, pitch and lead-lag are in phase but 90 degrees out of phase with flap.

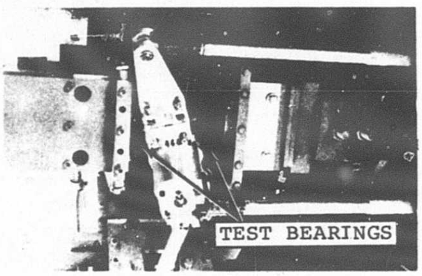

Figure 22. Fatigue Test Machine.

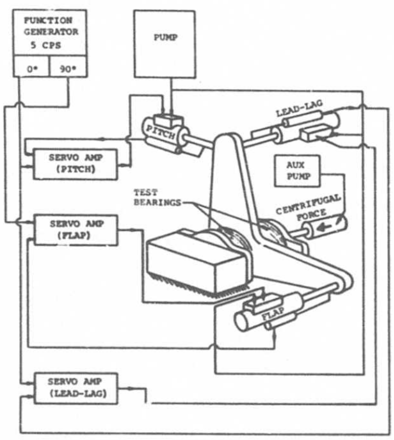

Figure 23. Schematics of Test Machine.

ENDURANCE TESTS

All endurance tests were monitored by periodic recordings of the spring rates and external conditions of the bearings. A pair of early test bearings with known mechanical damage to the steel shim flanges, resulting in a measurable 1.0-inch by 1.5-inch void in the rubber on one of the bearings at 400 hours and visible abrasion of rubber (fine hairlike pieces of rubber), were deliberately permitted to complete the 1200-hour endurance test. Figure 24 illustrates the consecutive deterioration of one of the bearings in association with corresponding changes to the spring rate in compression. For comparison, similar curves for a sound bearing of the final design are also shown.

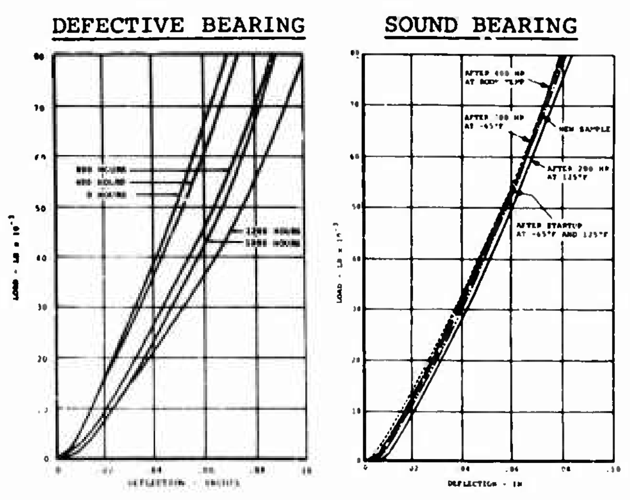

Figure 24. Compression Spring Rates per Bearing During Endurance Tests.

Figure 25 shows the lead-lag spring rates over 1200 hours of fatigue testing at room temperature. No significant change in spring rates can be detected. Even with the initial damage, the test samples successfully completed 1200 hours of endurance testing. This test has demonstrated fully the capability of this type of design to perform satisfactorily for long periods of time even with damaged sections.

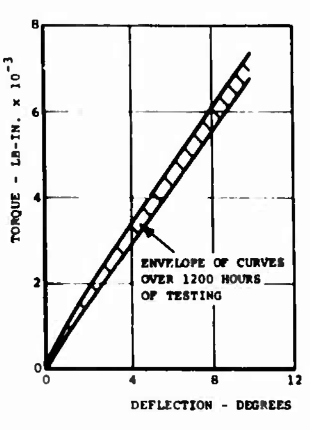

Figure 25. Lead-Lag Shear Spring Rate for a Pair of Bearings.

At the start of testing, temperatures were monitored carefully with thermocouples buried in the small end plate at the center of each part. The heat sink available in the test machine was not equivalent to the rotor head environment since an air stream is always present in the aircraft. Therefore, three small fans were added to the test setup. To demonstrate that hysteresis heat buildup in the bearings was not a problem, a test was performed with the blowers off. The temperature (blowers off) stabilized at approximately 106°F on a 77°F day, as was shown in Figure 21.

To conclude testing on test samples 1 and 2, the latter was used to perform 1000 cycles of repeated loadings in compression (CF direction) from 0 to 67,000 pounds to represent ground-to-air cycles. No change in the condition of the part was noted at the conclusion of the test. The same sample was loaded in torsional pitch (with no centrifugal force applied) to failure. The failure occurred at a torque of 20,000 inch-pounds and at a deflection of 118 degrees.

ENVIRONMENTAL ENDURANCE TESTS

Environmental endurance tests were performed on a total of four bearings, following the schedule in Table IV for samples 3 and 4. Typical results are shown in Figures 24 and 26.

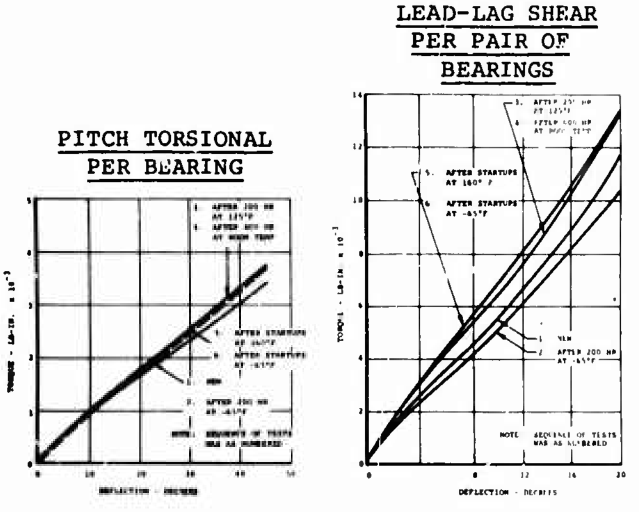

Figure 26. Pitch Torsional and Lead-Lag Shear Spring Rates During Endurance Tests.

INCREASED CENTRIFUGAL FORCE TESTS

In view of the gratifying results from preceding tests, two test bearings were subjected to endurance testing at 39 percent increased centrifugal force. The bearings were able to withstand this overload for 321 hours, at which time excessive abrasion of the rubber and cracks in the metal shims became apparent. The test was discontinued, and the bearings were deflected in pitch until failure. One specimen failed at a 130-degree wind-up with a torque of 25,000 inch-pounds. The other bearing was loaded in pitch to 25,000 inch-pounds with no failure.

MISCELLANEOUS TESTS

The effect of temperature on static spring rates in lead-lag, pitch, and compression is shown in Figures 27 and 28. The values apply to a pair of bearings simultaneously tested in the test fixture and therefore represent the sum of the two individual spring rates.

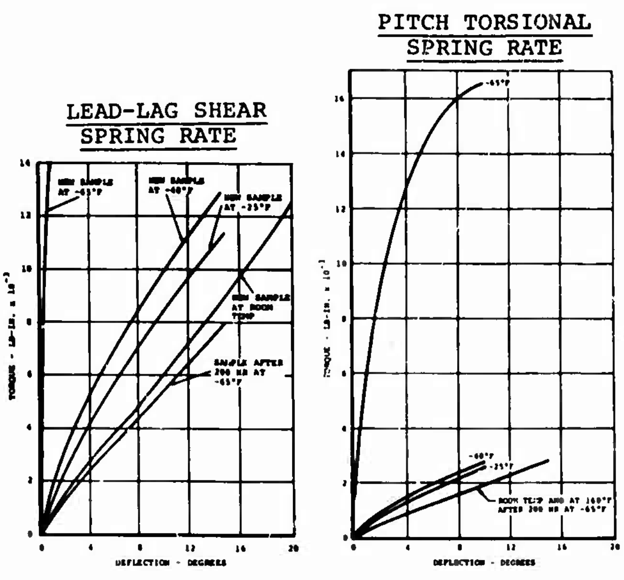

Figure 27. Lead-Lag Shear and Pitch Torsional Spring Rates per Pair of Bearings at Various Temperatures.

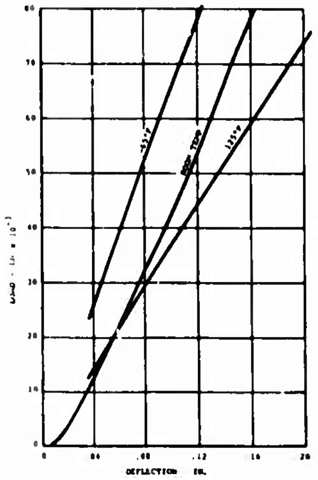

Figure 28. Compression Spring Rates per Pair of Bearings at Various Temperatures.

Figure 29 shows that a simultaneous constant deflection in shear (flap or lead-lag) has an effect on the pitch spring rate. The test was performed at room temperature.

It was found in other tests that the centrifugal force has practically no effect on individual spring rates.

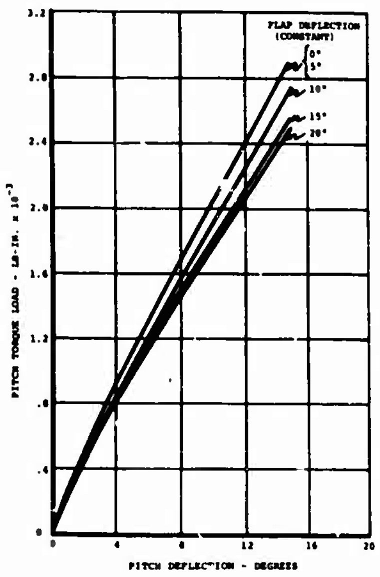

Figure 29. Combined Torsional Spring Rate (Shear and Pitch) per Pair of Bearings.

Dynamic tests were performed at low temperatures and very low frequencies to determine whether any reduction of stiffness can be expected if the pilot cycles the controls ± 10 degrees in pitch on the ground at 3 cpm for 10 cycles. At -25°F and -40°F there is no change in the torque required, and at -65°F the required torque is reduced about 25 percent (see Figure 30). Note that low-frequency cycling at -65°F required considerably less torque than the dynamic torque at -65°F for cycling frequencies in the normal operating range.

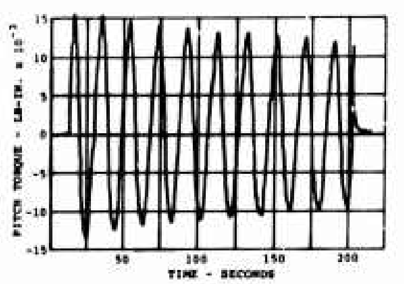

Figure 30. Dynamic Pitch Torque During First Ten Cycles of ± 10 Degrees Pitch at .05 CPS at -65°F.

To simulate the start-up of a rotor at -65°F, the bearing was deflected to full lead position (-15°), then soaked at -65°F until temperature stabilization. Instantaneous torque of 30,000 inch-pounds was applied and held constant for 30 seconds, then increased to 60,000 inch-pounds, deflecting the bearing to its full lag position (15°) over a total period of 40 seconds.

Absence of centrifugal force in this test introduced a high degree of conservatism, as the induced tensile stresses in the rubber and the bond were not compensated by compressive stresses. The test was repeated 100 times with no damage to the bearings.

CONCLUDING REMARKS

The spherical laminated elastomeric bearing is a structurally and functionally sound, completely practical concept.

The use of the elastomeric bearing in an articulated rotor system is a natural application to satisfy the functions of such a rotor hub. The hub is equally suitable for pure and compound helicopters.

Additional development work on structural components of the hub will have to be pursued to match the high degree of inherent fail-safety of the elastomeric bearing.

The predicted dynamic behavior of the elastomeric-bearing hub concept will have to be verified by a whirl test.

Successful development of an elastomeric-bearing rotor hub, combined with parallel programs on advanced-geometry blades using composite structures, is a step towards the ultimate goal of a safe, simple, reliable, and efficient rotor system of the future.

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ELASTOMERIC BEARINGS FOR UH-2 HELICOPTER TAIL ROTOR ARTICULATION

by

Paul F. Maloney
Kaman Aircraft

The subject of this discussion is elastomeric bearings that were designed for replacement of the UH-2 tail rotor flapping and pitch bearings. They are to replace typical rolling element bearings. The intent in initiating this research program was to devise a dry rotor head requiring no lubrication maintenance. All elastomeric bearings in this program were designed, fabricated, and tested by the Kaman Aerospace Corporation.

Figure 1 depicts the aircraft that the design is intended for, the Navy's UH-2, Seasprite, a standard utility helicopter. This happens to be the UH-2C that has the twin T-58 power on either side of the pylon. The tail rotor, which is where the bearings under discussion are located, turns 1720 rpm and produces a centrifugal force of about 20,000 pounds. The bearings that we have designed have to carry that load.

Figure 2 is a detailed view of the tail rotor. The upper sketch is a view looking down the shaft axis. The three hub lugs are integral parts of the rotor shaft which accept the flapping pins. Flapping bearings are mounted on the projecting ends of the pins. A pitch bearing is located in the housing retaining the blade shank. Since this rotor is essentially a zero delta 3, the pitch bearing moves only when the pilot provides control input. It has no one-per-rev type of motion. However, the flapping bearings do normally have one-per-rev flapping motion. The 20,000-pound centrifugal force is shared by the two flapping bearings. Figure 2 shows the lubrication provisions for the standard pitch bearings. There are also lubrication provisions in the end caps for the standard flapping bearings. The grip is attached to the hub with a through flapping pin. The bearings are assembled into the grip and slipped over the hub lug, after which the pin goes through the bearings and lug and the caps go on. The rotor, as shown, has freedom about the flapping axis and has no lead-lag hinge. However, it does have a limited amount of in-plane freedom by virtue of the fact that the through pin has what we call a "rocking" surface, which has a large radius. It provides limited freedom for the blades to swing

in plane; in fact, that feature is used as a tuning device in order to minimize blade loads.

It is pertinent to note that this development work was originated by Kaman as a company project and then was sponsored by the Navy in later phases. Prior to this, the Navy sponsored an ECP action for improved rolling element bearings for these positions, and so that development was going on while the elastomeric bearings were being developed.

Figure 3 shows the section through the housing illustrating the pitch bearing design. The blade spar, which is a forging, has an integral shank and flange at the inboard end, and the standard two-row pitch bearing which goes in that space is a split bearing. It is assembled around the shank, and then that assembly is slipped into the housing and secured by the large nut. The goal of this design effort was to simply replace bearings without redesigning any of the basic parts of the rotor. Consequently, these bearings had to do the same thing that the ball bearing does; that is, it had to be assembled around the shank. The metal rings have a double split so that they can be assembled around the shank. The elastomeric bearing, which is made of five individual stacks with intermediate spacers, has a single radial split so that the stacks can be slipped over the blade shank and into position as shown. Some preliminary work indicated that there was a stability problem with a bearing of this length under the design loads, and consequently lateral supports were added to handle the tendency to buckle. In addition to centrifugal forces, we must transmit the bending moments primarily edgewise or in-plane. This is carried as differential pressure through the pitch bearing; however, for extremely heavy inputs or maneuver type operations, a secondary surface was provided that the blade could contact. The normal path for shear loads is directly through rings 2 and 3 of Figure 3, which are always in contact. Under extreme loads, ring 4 also contacts the blade shank.

Since there was considerable constraint on the design, the proportions of steel and rubber were very carefully chosen. Figure 3 shows the breakdown of what was actually used for one of the five stacks. The two face laminations were 5-mil steel; the rest of the stack consisted of 35 laminations of 2-mil steel and 36 laminations of 2-1/2-mil silicone rubber. Steel was stainless type 302. The problem of control at -65°F led to the choice of silicone rubber for the pitch bearing, the concern being, of course, that the pilot could very readily, in extremely low temperatures, lift the ship off and then find that his pedals were extremely stiff or possibly even uncontrollable.

Figure 4 is taken from an Air Force report. Since the pitch bearing had a particular problem in terms of low-temperature performance, we

selected silicone rubber for that position. However, for the flapping bearing, because of the need for high strain performance and also the fact that the flapping bearing would feel only rotor forces and that there was no specific load that the pilot had to overcome, natural rubber was chosen for the flapping bearing.

Figure 5 shows the result of a test for a single stack of the pitch bearing which had a single split (omitted from the sketch). A rather broad hysteresis loop, which seems to be a fairly basic characteristic for the silicone rubber, can be seen. Also, the test was run for two values of axial compressive load - 2,000 pounds and 16,000 pounds - which approximated the rotor centrifugal force. It was conducted at room temperature. This type of bearing is seen to be relatively insensitive to the amount of axial load or thrust load applied.

Figure 6 shows the results of the bench endurance testing for the pitch bearing as well as the targets that were design objectives the bearing was to meet. The full pitch range of the tail rotor is 30 degrees from stop to stop or ± 15 degrees maximum vibratory. Very few cycles of that magnitude will be encountered in a typical helicopter. The ship is equipped with an ASE for the directional system, and therefore normal steady flight encounters relatively little vibratory pitch motion. Two degrees at 20 cycles per minute had been selected as the typical level that the pilot might do as a continual type of correction. Actually, from steady-state flight data, the normal pitch motion is much less than that; however, two degrees was considered to be achievable, and it was chosen as a goal. A series of endurance tests was conducted as shown in Figure 6. The first test, P_1, ran at ± 15 degrees for 14,200 cycles, at which time it was considered to have failed. The second specimen was run at ± 2 degrees to the goal with no apparent failure of any kind. That bearing was then retested at the ± 15-degree level and sustained failure at 6300 cycles. Then the last bearing, P_4, was tested at ± 5 degrees of pitch. It ran out to 274,000 cycles with only a slight indication of deterioration. It is therefore labeled as a termination since it could have gone well beyond that point. The dashed line in Figure 6 is the estimate of what silicone rubber alone should have been capable of, and the solid line indicates the approximate performance of the actual bearings as designed. Tests P_2, P_3, and P_4 included the simultaneous application of vibratory edgewise moment in order that this effect, which also produces fatigue loads on the bearing, would be fully accounted for in the results.

Figure 7 depicts the installation of the flapping bearing. Primary load, once again, is centrifugal force; however, the superimposed vibratory edgewise moment which was reacted through these bearings on the flapping

pin produces a vibratory radial load on the bearing, which is a possible additional source of distress. This configuration with a dual split was the result of preliminary tests of several configurations. Details of the bearing are shown: 33 laminations of 2-mil steel and 34 laminations of 4-mil natural rubber, giving a total thickness for the actual laminate of 0.202 inch. The inner and outer races have a thickness of 0.039 inch. Due to design constrictions, the approach was to achieve the maximum amount of rubber in the available space; consequently, the amount of steel was minimal.

Figure 8 shows the results of a stiffness test of flapping bearings. This bearing is of natural rubber; therefore, a fairly strong influence of temperature was anticipated, but it did not really appear with this data. The -20°F curve was actually an attempt to achieve -65°F, and the equipment that was available did indicate -65 degrees at one particular location. However, it was not possible to sustain it throughout the parts, and the most representative temperature was -20. Thus, even with this relatively broad range of temperatures, we show relatively little change in stiffness due to temperature; also, in comparison with silicone, we showed very little effect of hysteresis. The test setup has four bearings on a single shaft, with two of them loaded and the other two reacting the load. Therefore, the four bearings have equal radial load; and the torque necessary to rotate the shaft is divided by four, so that Figure 8 presents the average of four bearings.

The flapping bearing or any radial bearing has some kind of a loading distribution associated with it. Now, from several kinds of observations, we conclude that the load distribution is approximately as shown in Figure 9. It may not be precisely uniform throughout the central area; it may be something like a cosine distribution, but the most important thing is that there is a relatively steep gradient in line with the projected sides of the inner race. This conclusion comes from observations of loaded bearings and also failed parts. With this type of a load distribution, the laminate that moves from point 2 to point 1 in Figure 9 is moving through an area where this load is varying rapidly, and this was evidenced in our fatigue tests.

Figure 10 is a three-dimensional way of looking at performance of these radial bearings, which were quite highly loaded. The upper sketch, I, illustrates a failure surface, assuming rubber to be the limiting factor. It defines the performance of the rubber. The vertical axis is $\pm \Theta$, the angle of oscillation that the bearing goes through. The horizontal axis to the right is the number of cycles of oscillation; the horizontal axis to the left is the radial force or the steady load applied to the bearings, so that

at any given steady load, a section through the surface produces an S-N curve, in the sense of angle oscillation versus the number of cycles, considering only the performance of rubber. Now superimposed on this, we found, as have others, that metal laminate fatigue could be a possibility. Going back to Figure 9 and considering the variation in the local stress under the load distribution in the bearing, it is apparent that a steady load will produce vibratory stresses during applied cycles of oscillatory motion. The result will be an S-N curve shape for the metal laminates of the type shown in sketch II of Figure 10. The steady radial load on the bearing produces tension stresses within the laminates, and in the area where the load is changing rapidly these stresses are highly biaxial. The magnitude of these stresses is a function of the steady load. Sketch III of Figure 10 illustrates a superposition of these effects, showing a scooped-out section of the rubber performance curve (or performance surface) as influenced by the performance of the metal laminates themselves. You could, of course, move this out or possibly eliminate it as a source of trouble by having relatively thick metal laminations compared to the amount of rubber. But that would compromise the bearing design. Given more design freedom, it would be pertinent to change the proportions of metal to rubber, but that was not possible in this program. You will notice that a section through the surface at high load levels produces a two-part S-N curve for the bearing, the low-cycle part defined by the rubber and the high-cycle part defined by the metal.

Figure 11 illustrates such a curve and also shows the results of the bench fatigue tests of the flapping bearings. Square targets indicate the selected design objectives, which it is apparent were not fully met. The full flapping motion from stop to stop is ± 10°. That will not occur too often. Steady, high-speed, level-flight flapping is between 2° and 4°. We had selected 4-1/2° as a target point, which, if achieved, would provide a bearing that would have essentially no damage in any steady, level-flight condition.

Preliminary testing of a prototype specimen yielded point F01 on Figure 11, which was a nonfailure. The same bearings were subsequently run at ± 4° to point F02. The point is shown with an arrow because the failure was associated with a manufacturing defect in the bearing. These specimens were produced when the laboratory was learning how to make these bearings; however, they were considered to be of adequate quality to justify testing.

Specimen F1 was then tested at ± 5° and produced a complex mode of failure of both metal and rubber. Based on observation of that specimen, it was considered possible that the metal might have been very influential in the failure.

Specimen F2 was run at ± 8° in an attempt to develop S-N data. It was run to 70,000 cycles at that level, and on inspection there was found to be deterioration at the site of a minute manufacturing defect. Further testing of that bearing was considered to be inappropriate.

Test F3 was initiated at ± 3-1/2° with very careful monitoring to identify the very first indication of failure. With the exception of some extremely minute rubber fuzz all over the bearing, the very first sign of the failure was one laminate coming out of the center of the stack. This fracture occurred at the location of high load variation that is indicated on the load distribution curve, Figure 9.

The last test was at 6°, and it was run to 40,000 cycles. At that time, the bearing was still functioning; however, close inspection revealed that some rubber was extruding and that a steel laminate had cracked. The original judgement was that rubber should have been capable of operating to the upper dashed line. The bearing as designed was expected to operate to the intermediate line; however, a limitation in the form of steel failure was encountered. Failures in the level above 6° were associated with the rubber, with no sign of any steel problems; hence, it must be concluded that there is a compound type of failure mode.

Improved performance would be attainable by a choice of better material for metal laminates. Stainless steel type 302, full hard, was used; it had a tensile strength of about 220,000 psi, which is not extremely high for the very thin laminations. An alternate approach would be to sacrifice some rubber to get more steel into each lamination. Obviously, given complete freedom to redesign the basic components, an optimum bearing with better performance could be provided.

A prior program for the development of rolling element bearings primarily for the flapping axis but also for the pitch axis was successfully completed during the course of the elastomeric bearing program. The bearings resulting from that program were a primary consideration in terminating our efforts on elastomeric bearings for the UH-2 tail rotor.

It can be concluded that a satisfactory design for the pitch bearing was developed; however, further developmental work is necessary before flapping bearings can achieve their full life potential.

Figure 1. UH-2C Seasprite.

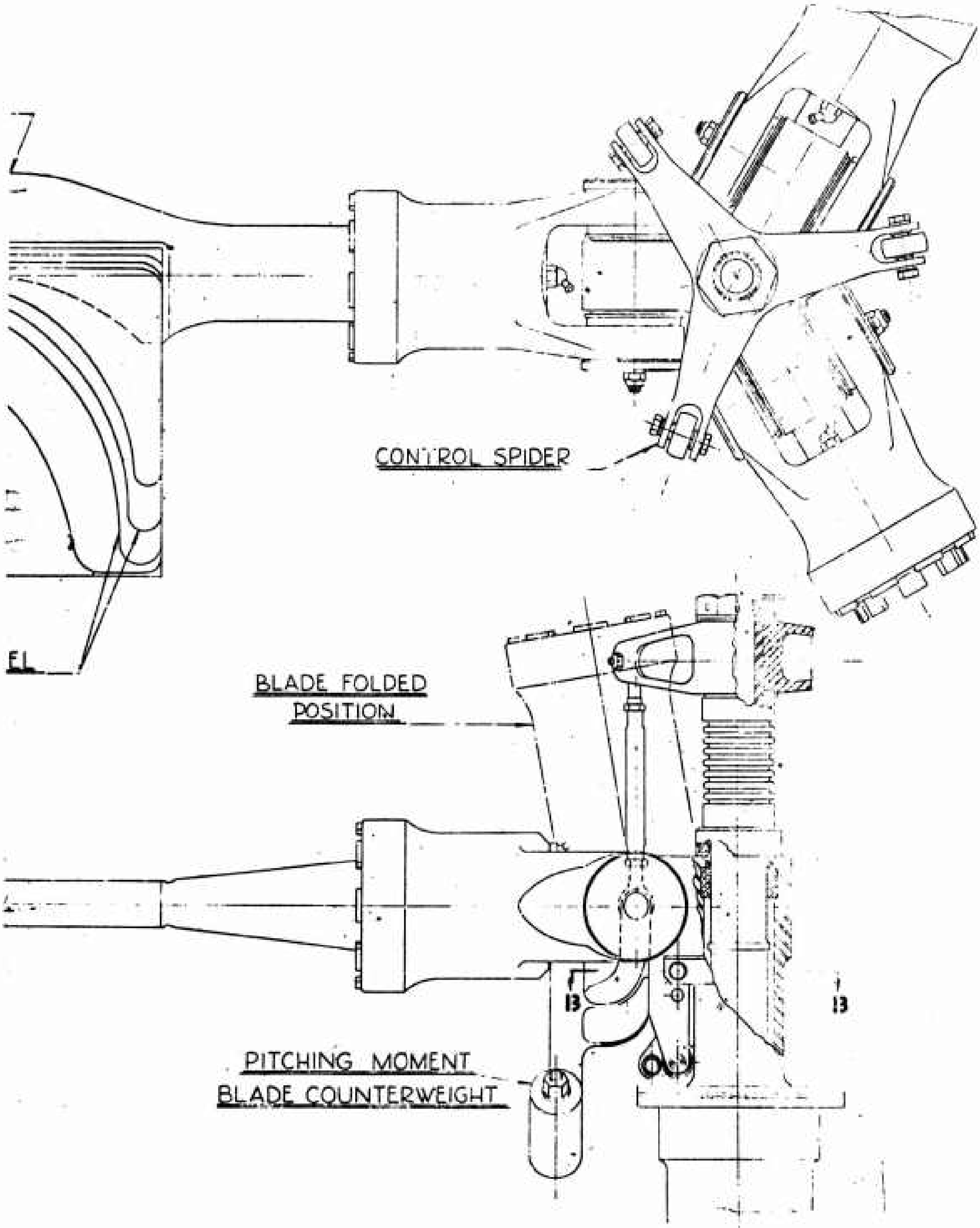
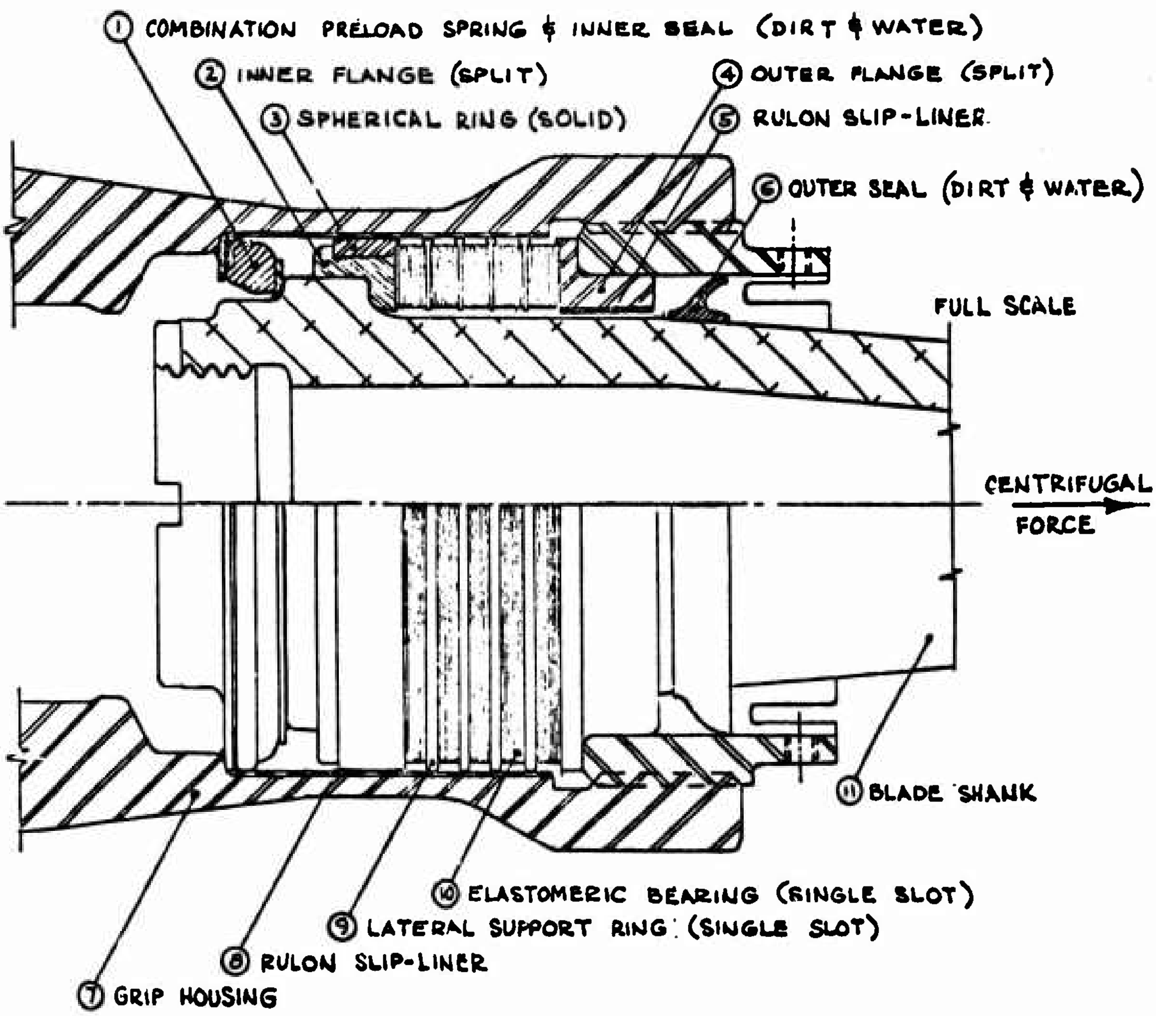


Figure 2. Detailed View of Tail Rotor.



2	LAMINATIONS	5 MIL STEEL (FACINGS)	-	.010
35	LAMINATIONS	2 MIL STEEL	-	.070
36	LAMINATIONS	2.5 MIL SILICONE RUBBER	-	.090
		INDIVIDUAL STACK THICKNESS		.170
		5 STACKS USED PER BLADE		

STAINLESS STEEL SHIM STOCK .... Type 302
2 MIL THICK - SPRING TEMPERED *

GENERAL ELECTRIC SILICONE PRIMER .... SS 4038

GENERAL ELECTRIC SILICONE RUBBER .... XE 508

Figure 3. Elastomeric Pitch Bearing Assembly.

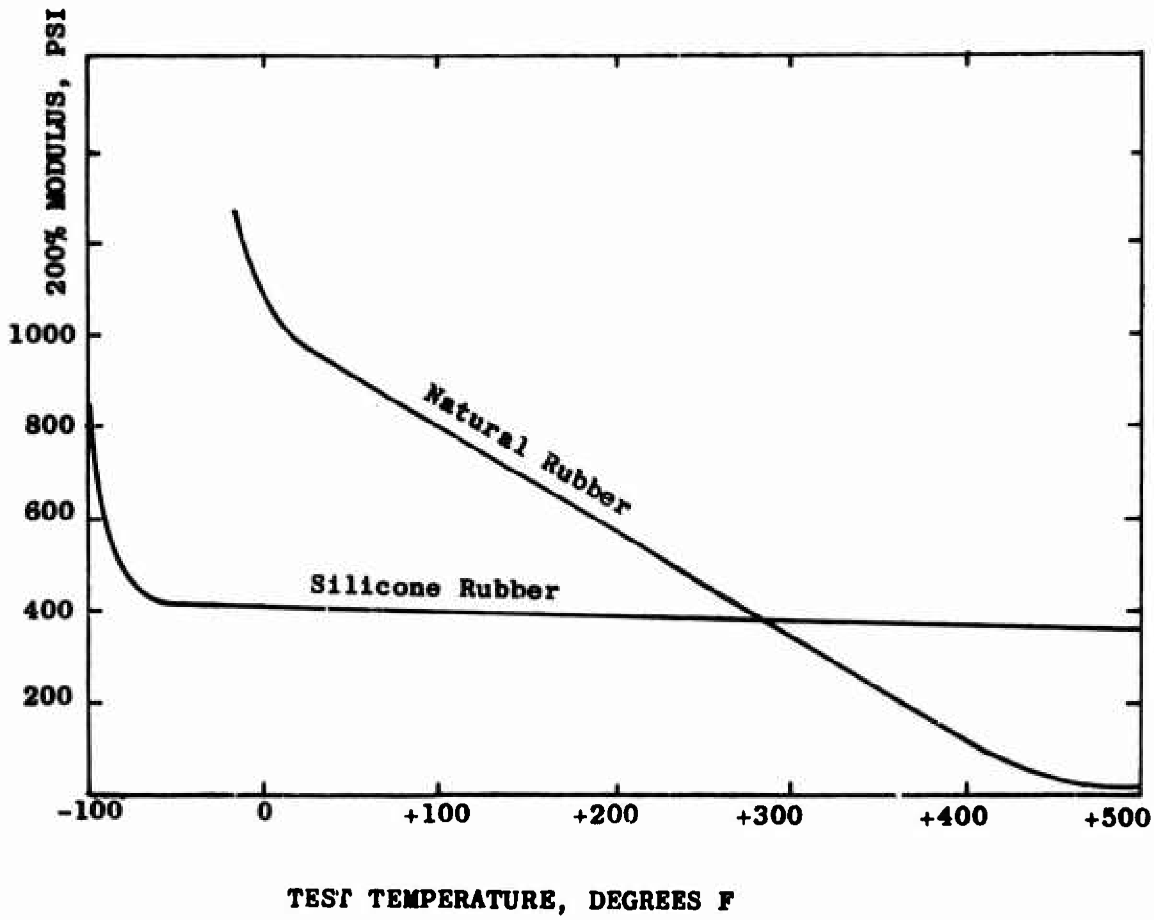


Figure 4. Modulus Versus Temperature.

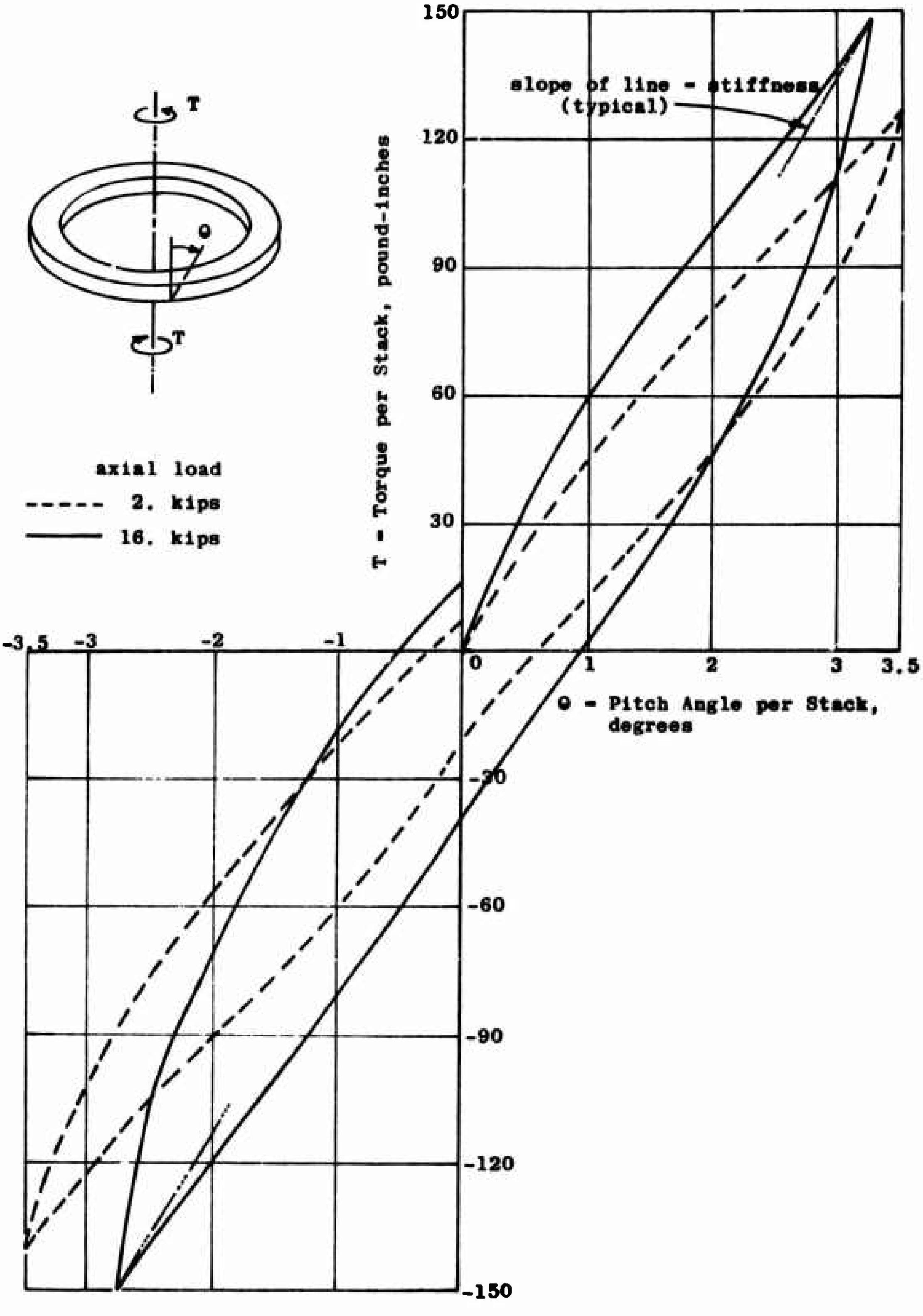


Figure 5. Pitch Bearing Stiffness, Single Stack, 80°F.

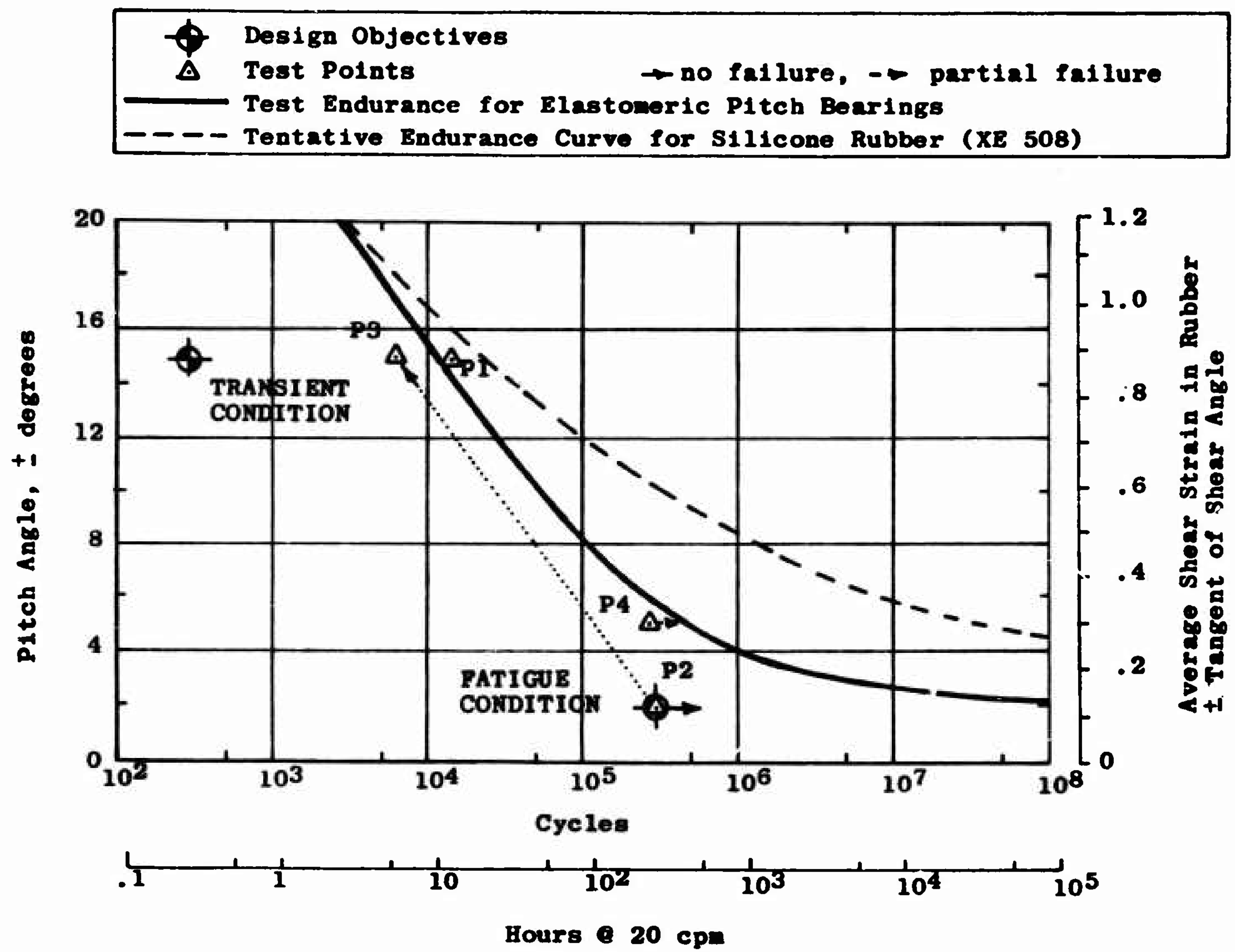


Figure 6. S-N Curve for Elastomeric Pitch Bearing.

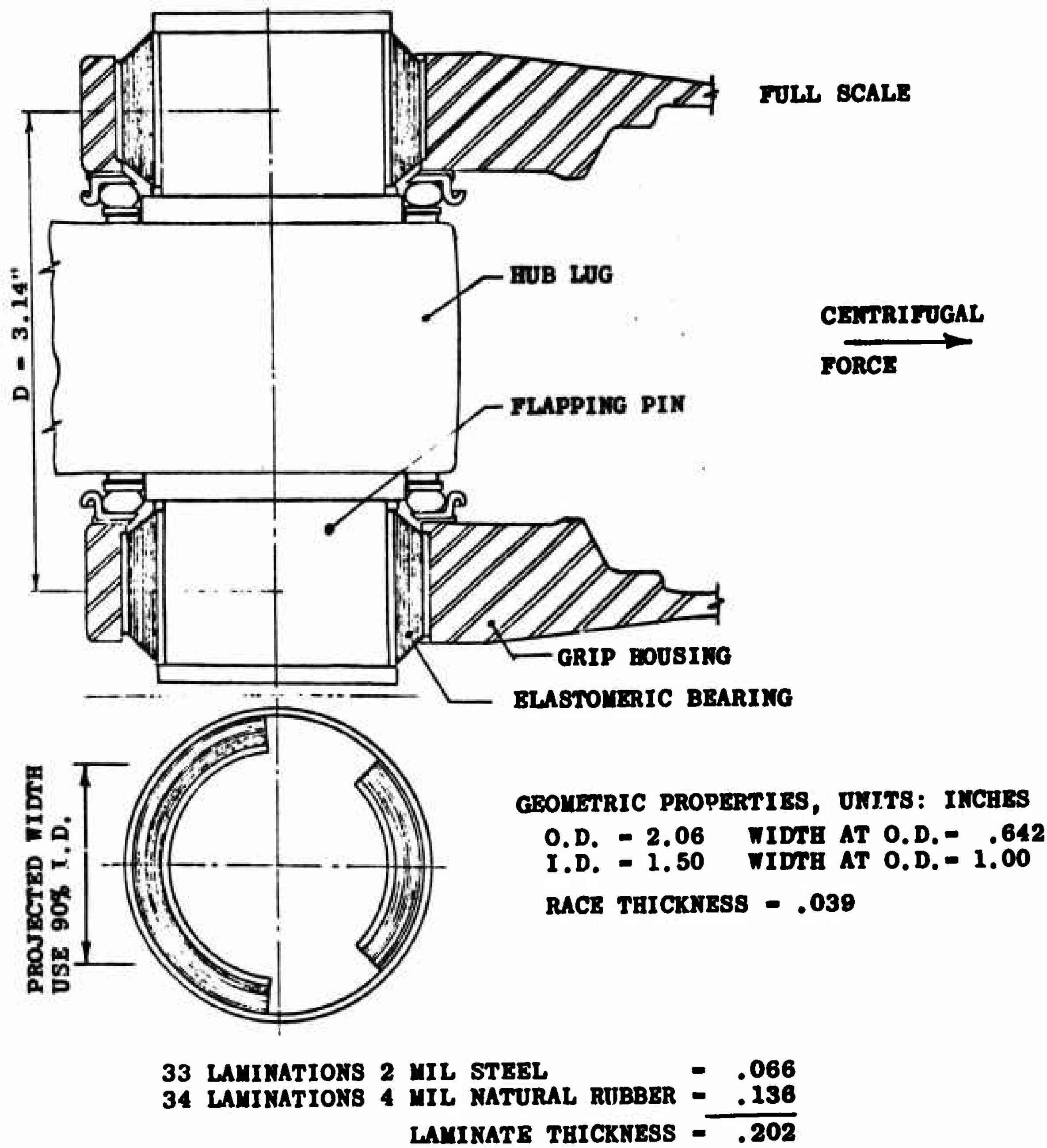


Figure 7. Elastomeric Flapping Bearing Assembly.

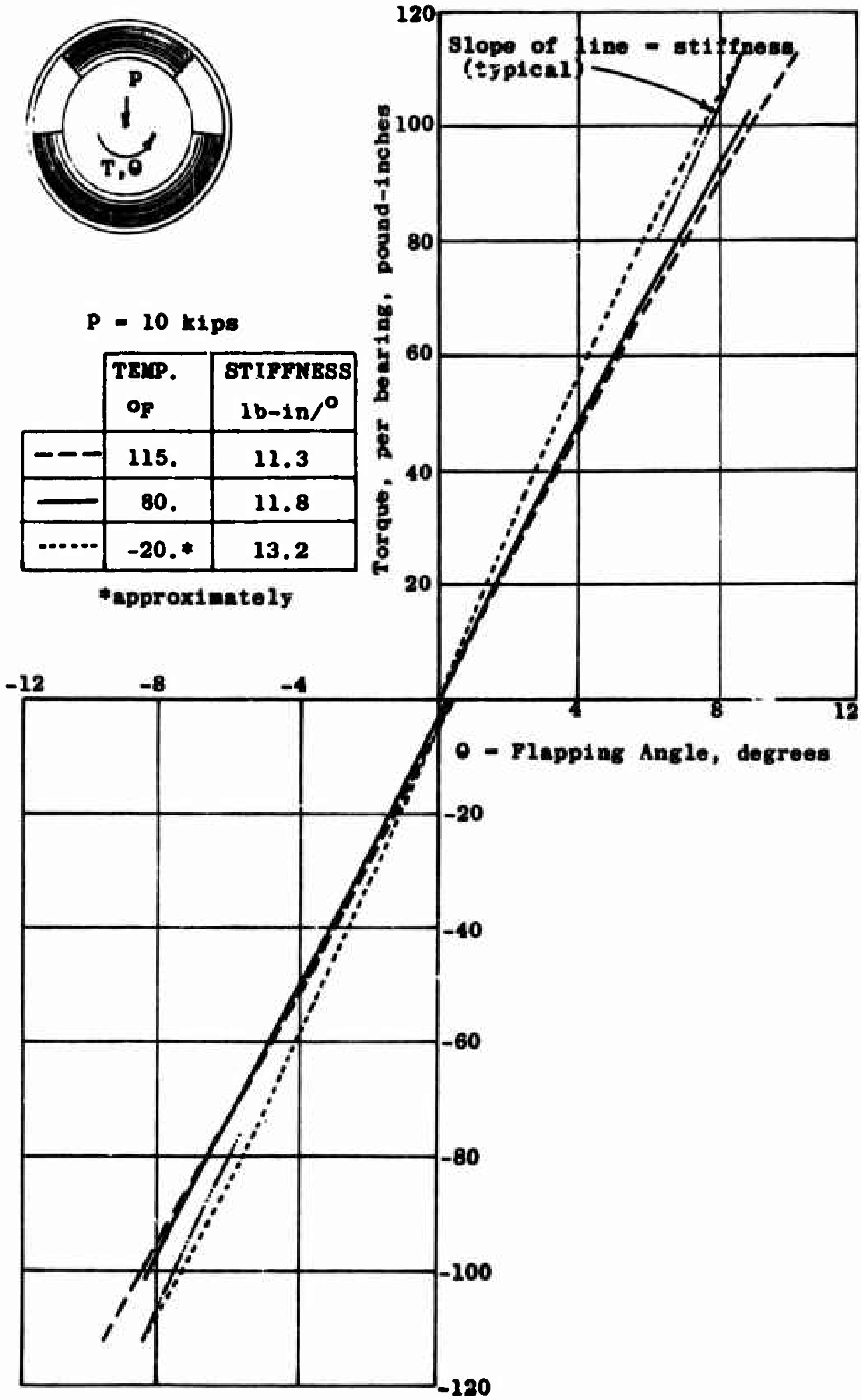


Figure 8. Flapping Bearing Stiffness, Single Bearing.

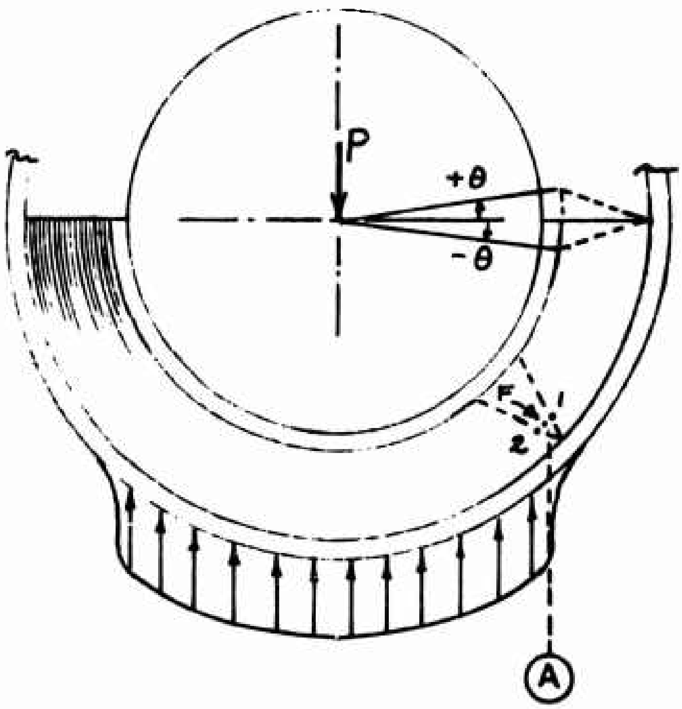


Figure 9. Representation of Load Distribution - Radial Elastomeric Bearing.

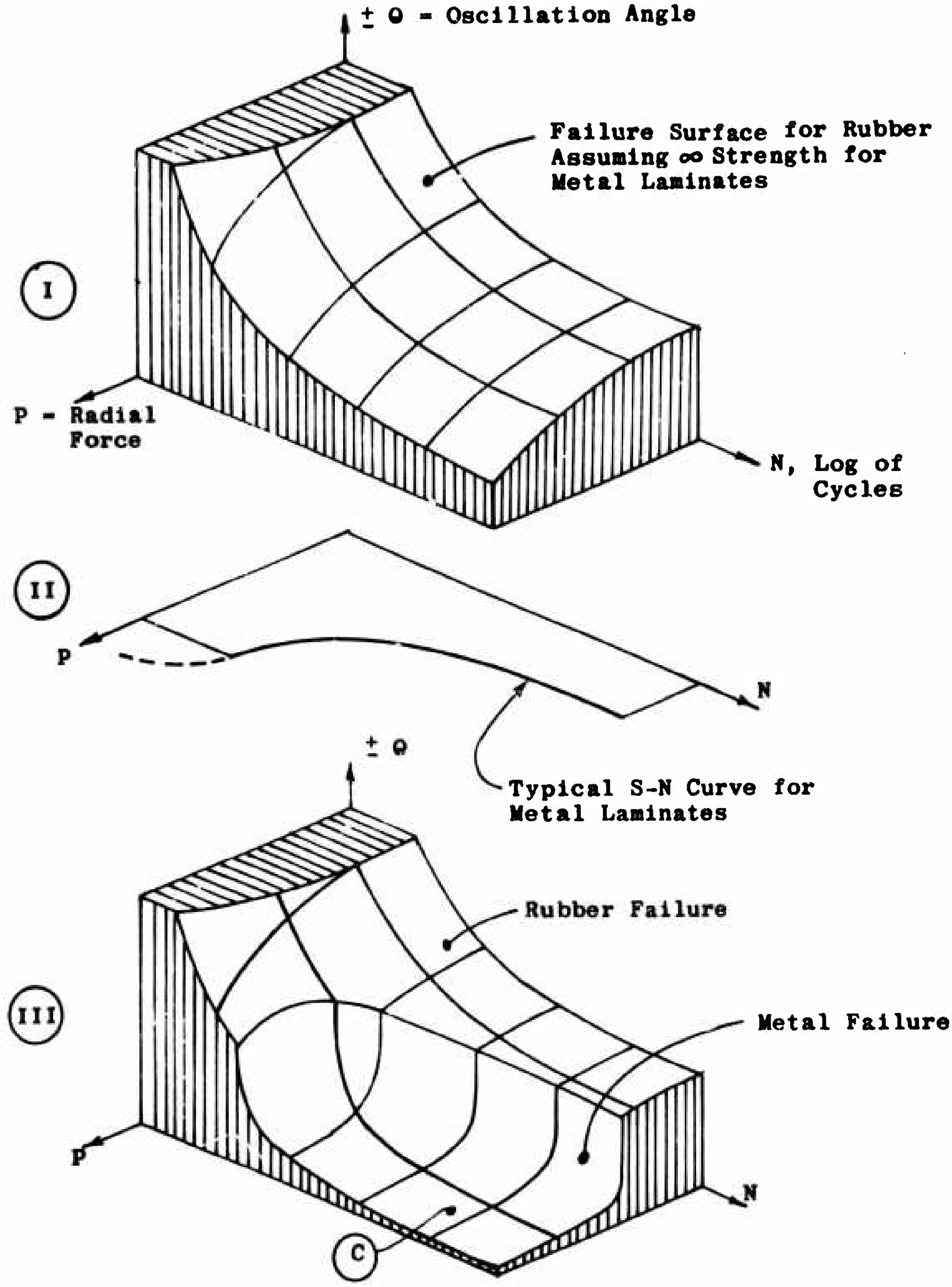


Figure 10. Three-Dimensional Θ-N-P Failure Surface for Radial Elastomeric Bearings.

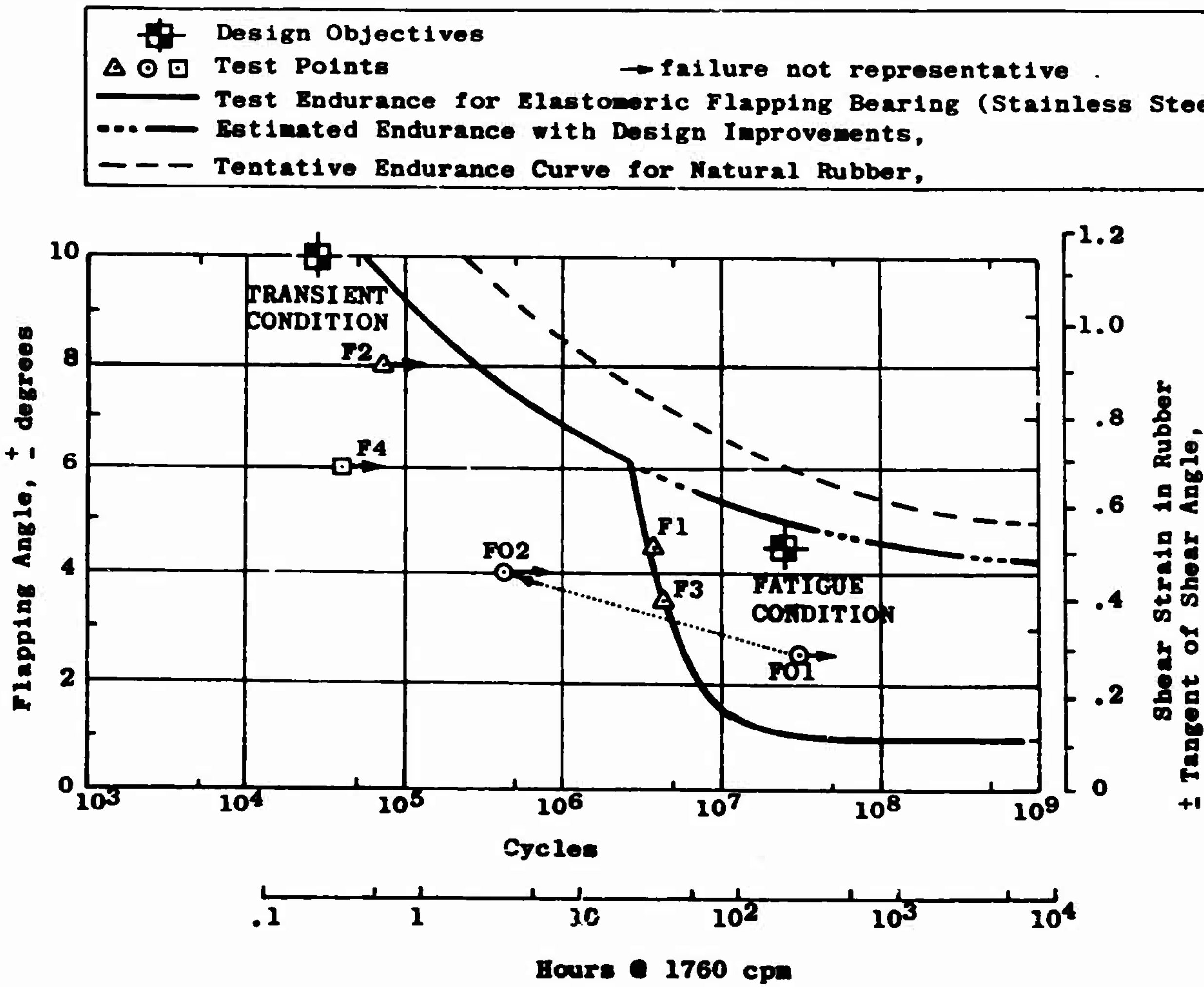


Figure 11. S-N Curve for Elastomeric Flapping Bearing.

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POTENTIAL OF ELASTOMERIC BEARINGS FOR ROTOR APPLICATIONS

by

C. H. Fagan
Bell Helicopter Company

Elastomeric bearings differ in many ways from the conventional roller, needle, and other bearings that are available for use on rotors, so it is natural to ask: why use elastomeric bearings? The answer (Figure 1) is that we expect them to lower costs, to reduce maintenance, to eliminate the need for lubrication, to increase safety, and to improve our control of natural frequencies. Costs are not reduced by the bearings themselves, but by their permitting us to simplify rotor design and reduce the tolerances of parts. Maintenance is reduced because elastomeric bearings wear gradually and visibly; they can be designed for visual inspection without disassembling the rotor. Operation without lubrication is an inherent advantage of elastomeric bearings. Safety is increased not only because these bearings are naturally fail-safe, but also because their wear is gradual and visible. Frequency control is important because the loads on tail rotors can be reduced by careful placement of their natural frequencies. Changing the stiffness of the blade-grip bearings is an important means of frequency control, and elastomeric bearings facilitate this tuning.

WHY ELASTOMERIC BEARINGS

- LOWER COST
- REDUCED MAINTENANCE
- NO LUBRICATION
- INCREASED SAFETY
- FREQUENCY CONTROL

Figure 1

At Bell Helicopter Company, we began to investigate the use of elastomeric bearings for rotors in 1964. Figure 2 lists some of the questions that have arisen. Manufacturing techniques are concerned with tolerances of the parts that retain the bearings and of the bearings themselves, for interchangeability. Design suitability refers to the weight and size of the bearings. Although these bearings are slightly heavier than conventional ones, a rotor that uses them is almost as light as a conventional rotor. The elastomeric bearings are generally somewhat larger, too, so the blade grip is also slightly larger than one using conventional bearings. The fatigue life of elastomeric bearings is considerably longer than that of conventional bearings in oscillatory applications. The questions of environment--temperature, humidity, and contamination--have been partially answered, but more work is necessary. Service life is probably the most important of these questions.

RELATED QUESTIONS

MANUFACTURING TECHNIQUES
- COST
- TOLERANCES

DESIGN SUITABILITY
- WEIGHT
- SIZE
- FATIGUE LIFE

ENVIRONMENT
- TEMPERATURE
- HUMIDITY
- OIL

SERVICE LIFE

Figure 2

Some of our attempts to answer these questions are listed in Figure 3. Both bonded and molded bearings have been investigated. The design and suitability of elastomeric bearings for tail rotor flapping and pitch-change have been evaluated. Their use in main rotors has also been studied, and fatigue tests have been conducted. The remainder of this discussion will be concerned with these four items.

Figure 4 shows a two-bladed, seesaw tail rotor. Elastomeric radial bearings are shown in the flapping axis, and thrust bearings are shown in the pitch-change axis. The blade grips are modified to accommodate the elastomeric thrust bearings, but the radial bearings are mounted in the same envelope as the needle bearings presently used in the UH-1 rotor. Three major tail-rotor programs, including flight tests, used

this type of UH-1 tail rotor--two under USAAVLABS contract and one as a part of Bell IR&D.

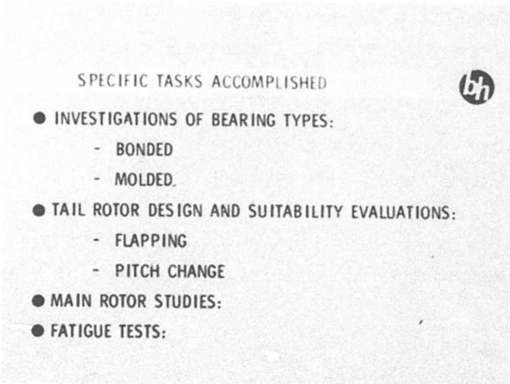

Figure 3

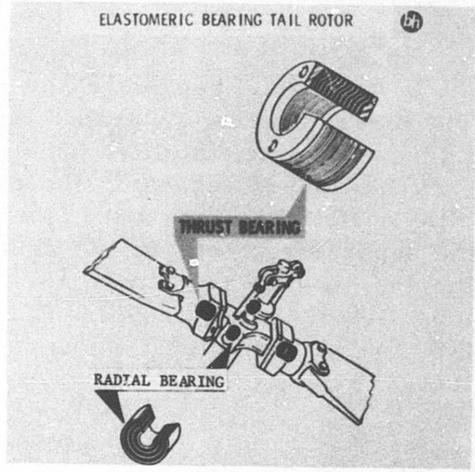

Figure 4

The first elastomeric bearings that we investigated (Figure 5) were of bonded construction. They consisted of thin layers of elastomer, 0.003 to 0.004 inch thick, with steel shims 0.002 inch thick. The radial bearings were fabricated by squeezing uncured elastomer on the shims, stacking the shims, and applying sufficient pressure to obtain the desired elastomer layer thickness. The uncured elastomer pad was rolled into a cylinder and inserted in the outer race. A split-cone inner race assembly was then inserted to apply pressure during the curing process. The thrust bearings were machined from cured elastomer pads.

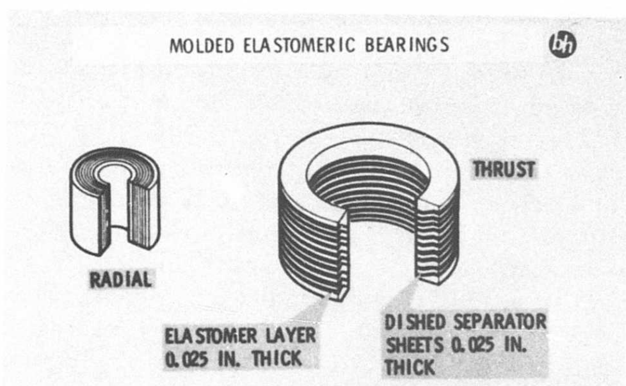

Figure 5

These bonded elastomeric bearings were fabricated at Bell in 1965. Some problems were encountered with the quality of these laboratory fabricated bearings, and there were some early failures of the radial bearings where the ends of the elastomer pads butted together. Therefore, the molded bearings shown in Figure 6 were investigated. Both the radial and the thrust bearings had thicker elastomer layers--approximately 0.025 inch thick--with steel separator sheets of approximately the same thickness. The thrust-bearing separator sheets were dished at 30 and 45 degrees, to improve column stability. One later configuration was made with a chevron-shaped separator sheet, which also increased the column stability. Both bearings were fabricated by the hot-mold transfer method: the separator sheets were stacked in a dye, elastomer was forced into the dye, and pressure was maintained during curing. The molded bearings, fabricated by Lord Manufacturing Co., were quite satisfactory in tail-rotor applications, so no further work was done on the bonded bearings.

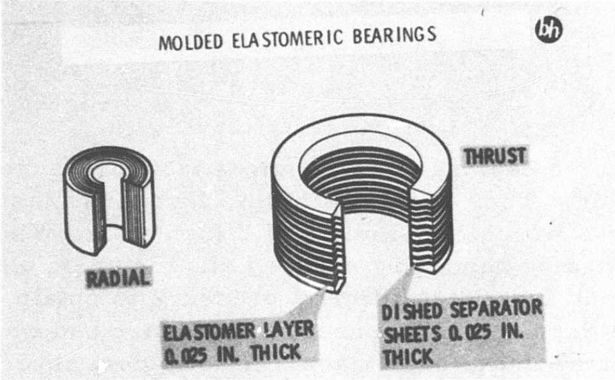

Figure 6

Design requirements for the bearings in a two-bladed, seesaw tail rotor are given in Figure 7. The radial flapping bearings are installed in a 35-degree delta-three hinge. Each one must accommodate a flapping angle of ± 8 degrees and a radial load of 570 pounds. The thrust bearing is designed for a blade centrifugal force of 16,000 pounds, and all the grip bearings must accommodate collective blade pitch from -6 degrees to +18 degrees. The thrust bearing in the early configuration had only to carry blade centrifugal force. In later configurations, however, it also transferred blade-bending loads to the yoke, resulting in a cocking-moment requirement of ± 2000 inch-pounds once per revolution, 1650 times per minute. This cocking moment creates a radial load requirement of ± 1000 pounds for the inboard grip (radial or journal) bearing.

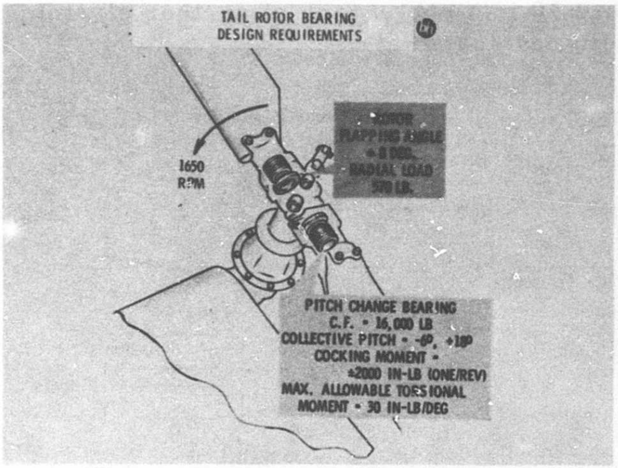

Figure 7

The maximum allowable torsional moment for all the bearings in one grip is 30 inch-pounds per degree. This includes the friction forces for conventional bearings if they are used in combination with elastomeric bearings. The torsional-moment limit is explained by Figure 8, which shows the tail-rotor-pedal control forces for a blade angle. The left ordinate shows the force the pilot must exert to move the pedals to the desired collective pitch setting. The crosshatched areas are unacceptable, and the space between them is acceptable. The No. 4 (desired) line represents normal operating conditions with the hydraulic boost system operating properly. The No. 1 line shows boost-off pedal forces for the UH-1; the No. 2 line, boost-off operation with the elastomeric bearings in the tail rotor. The No. 3 line shows the pedal forces required for the elastomeric-bearing rotor during boost-off operation at -65°F. For example, let us assume that the pilot wants to make a maneuver which requires a +12-degree tail-rotor-blade pitch. In normal operation with any tail rotor, a left-pedal force of approximately 20 pounds is required. For boost-off operation at normal temperature, however, the force is 40 pounds for the UH-1 rotor and 55 pounds for the rotor with elastomeric bearings. At -65°F, this 55-pound boost-off force increases to 65 pounds. Even though the elastomeric-bearing values are higher than those for conventional bearings, they are well within the acceptable range. The torsional stiffness of elastomeric bearings creates no insurmountable problems.

Figure 9 is a photograph of the all-elastomeric-bearing tail rotor mounted on a UH-1 tail-rotor gearbox. UH-1 blades were used, and the grips were modified for the elastomeric bearings. The radial flapping bearings were installed in line with the controls to prevent blade-pitch changes with

flapping. Figure 10 is a photograph of another elastomeric-bearing tail rotor that is being evaluated at Bell.

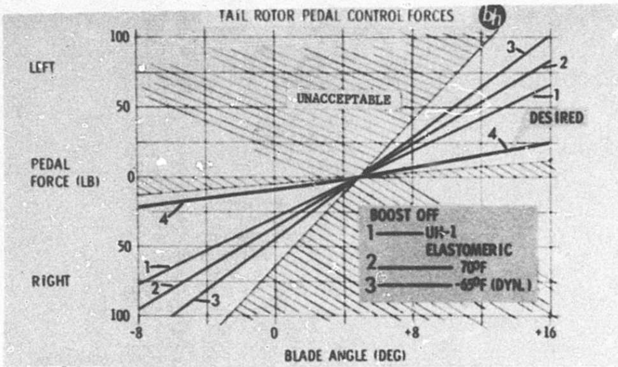

Figure 8

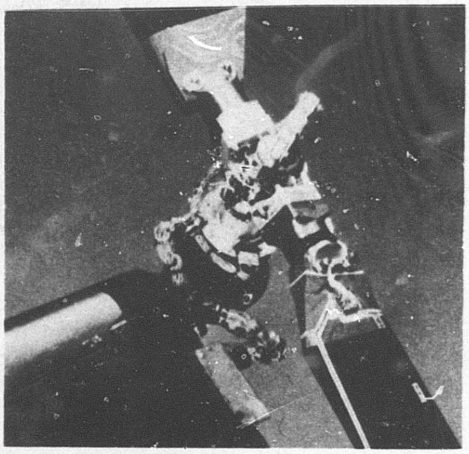

Figure 9

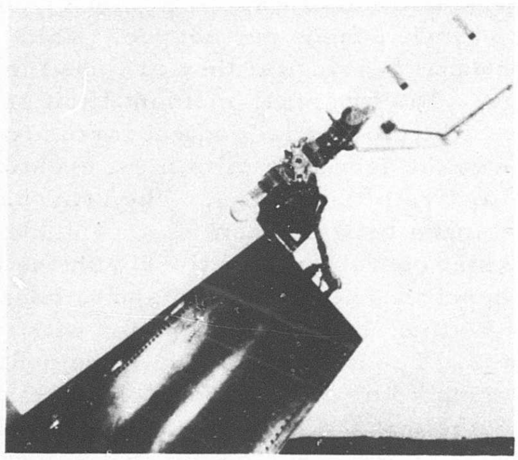

Figure 10

The tail-rotor configurations that were flight-tested with UH-1 blades fall into two groups, called "stiff" and "soft" grip configurations. Figure 11 is a sketch of a "stiff" grip in which two Teflon journal bearings transfer the blade-bending loads to the yoke, and an elastomeric thrust bearing carries the blade centrifugal force. The outboard end of the thrust bearing is pinned to the yoke, and the inboard end is pinned to the grip. Thus pitch change is accommodated by torsional deflection of the thrust bearing. One of the "soft" configurations was achieved by removing the outboard Teflon journal bearing, as shown in Figure 12. The thrust bearing

carries the blade centrifugal force, and it also transfers the blade-bending loads to the yoke. The journal bearing acts as a pivot and carries the blade-shear loads. The purpose of the "soft" grip configuration was to tune the rotor's natural frequency away from the helicopter's operating resonant frequencies.

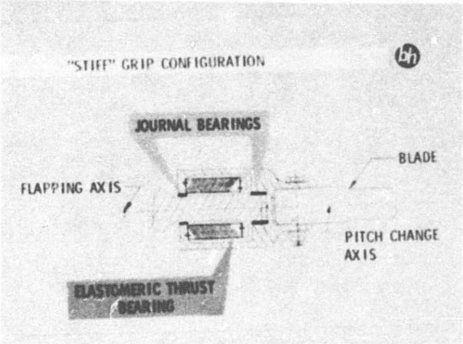

Figure 11

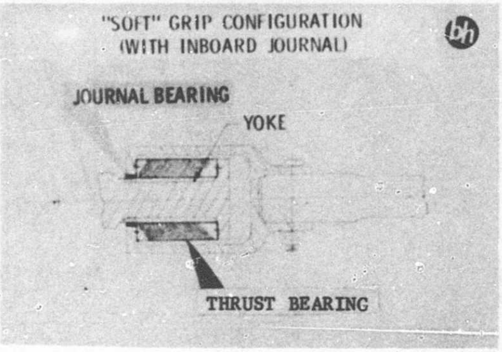

Figure 12

Figure 13 is a sketch of another "soft" configuration that was tested during the all-elastomeric-bearing tail rotor program. The thrust bearing carries the same loads as the previous configuration, but an elastomeric radial bearing in the grip carries blade-shear loads and acts as the pivot--instead of the journal bearing. Molded elastomeric radial bearings were used in the rotor flapping axis. Cocking motion is imposed on the thrust bearing during transfer of the blade-bending loads to the yoke as illustrated in Figure 14. Dynamic and static curves show the moments required to cock the two thrust bearings (loaded as shown in the sketch). The moments given are for two bearings; values for the grip would be only half of those given, since only one bearing per grip is loaded. The normal operating range for the "soft" grip is approximately ± 0.2 degree cocking 1650 times a minute. This motion is sufficient to keep the bearing warm, even during extremely low temperature operation, and prevent excessive bearing torsional spring forces. In fact, the torsional spring force increases by only 40 to 50 percent in operation at -65°F.

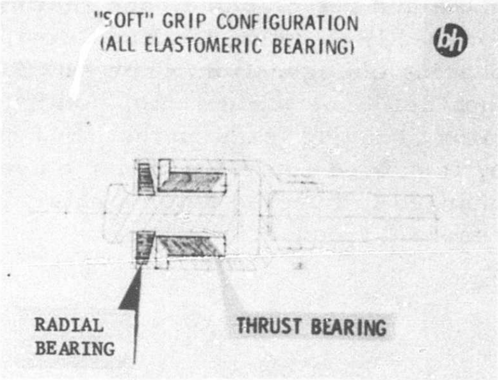

Figure 13

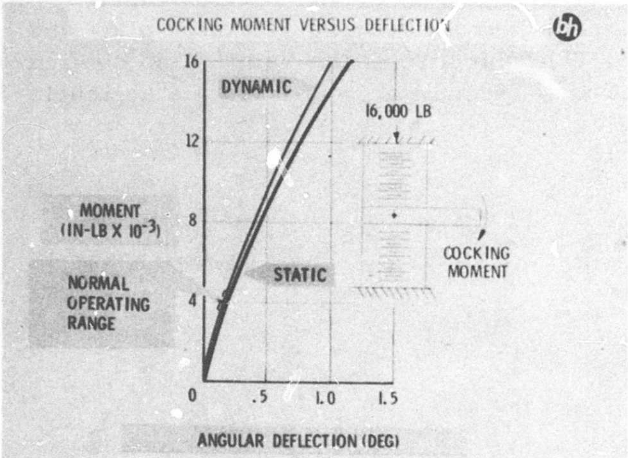

Figure 14

Figure 15 compares typical elastomeric-bearing flight-test results with data for the UH-1 tail rotor. Oscillatory blade-bending loads, both in-plane and out-of-plane, are shown. The No. 1 curve represents values for the UH-1 rotor; the No. 2 line, those for the "stiff" elastomeric-bearing configuration. The No. 3 line shows values from flight tests of the "soft", or tuned, rotor configuration. The main point is that oscillatory bending loads on the UH-1 rotor were reduced by 20 to 25 percent by improved rotor-frequency placement. Blade-to-yoke stiffness was changed to tune the rotor's natural frequency away from the operating resonant frequencies.

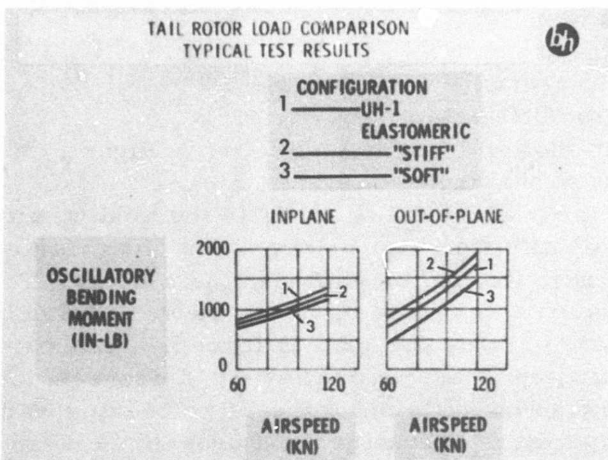

Figure 15

An all-elastomeric-bearing grip, designed to take advantage of the elastomeric bearing's favorable characteristics, is shown in Figure 16. The thrust bearing carries blade centrifugal force and bending loads. The inboard radial bearing prevents grip translation and carries the blade-shear loads. A window permits inspection of the thrust bearing; the visible gradual wear indicates the bearing's remaining service life. The radial bearing can be similarly inspected from the inboard end of the grip. After the service life of the thrust bearing is established, the inspection window can be removed. The inboard radial bearing's condition can then be used as a gage of the remaining service life of both bearings. In fact, it can serve to estimate the condition or remaining service life of all parts in the tail rotor, since the oscillatory loads applied to this bearing are also directed through the other critical parts of the rotor. This grip needs no lubrication, and the inboard radial bearing is the only seal necessary for the grip-to-yoke attachment. The elastomeric bearings are expected to have good service life, since all blade motions are accommodated by deformation of the elastomer. The radial flapping bearings in this rotor are of the same molded type that was flight-tested in the later programs. The simplicity of the design and the elimination of close-tolerance parts will make the initial cost of the rotor low.

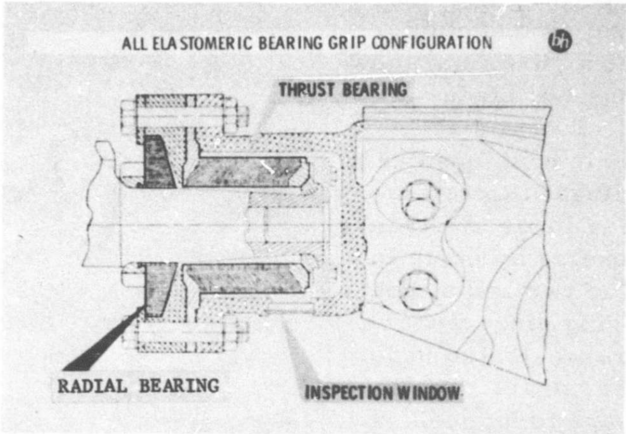

Figure 16

Advantages and disadvantages of the all-elastomeric-bearing rotor are compared with those of a conventional rotor in Figure 17. The proposed rotor has a slight disadvantage of weight, and a small cost advantage. Maintenance of the elastomeric-bearing rotor is estimated to be only half that of the standard rotor. The service life of the elastomeric bearings

is expected to be at least twice that of the conventional bearings. Furthermore, the conventional bearings require lubrication and seals that the proposed rotor does not need.

COMPARISON

	STD. ROTOR	E. B. ROTOR
WEIGHT	1	1.1
COST		0.9
MAINTENANCE REQ'D.		0.5
SERVICE LIFE	1	2.0
LUBRICATION	YES	NO

Figure 17

During the tail-rotor programs, main rotors were also studied. The resulting configuration, shown in Figure 18, has two blades and is stiff inplane. Two radial-thrust elastomeric bearings carry the blade centrifugal force, and the blade-bending loads are transferred to the yoke as a couple between them. The two radial bearings in the flapping hinge were made and endurance-tested under a USAAVSCOM contract. Their endurance appears to be good: by adjusting their size to the specific application, we can achieve a service life of several thousand hours. The radial-thrust bearings have not been fabricated; however, calculations show them to be satisfactory, including satisfying the blade-to-yoke stiffness requirements. No close-tolerance parts are required to retain the bearings. The use of elastomeric radial-thrust bearings eliminates the tension-torsion strap,

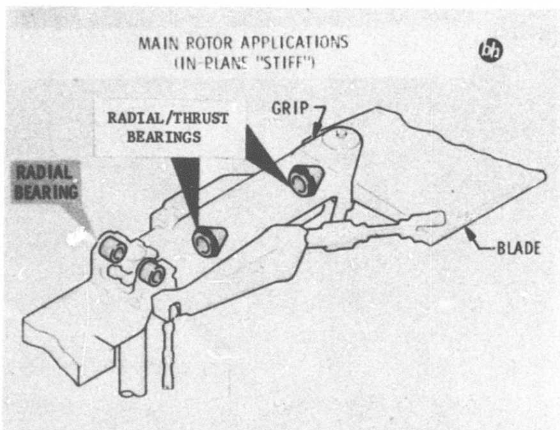

Figure 18

the conventional grip bearings, the oil retainers and seals, and a number of close-tolerance retaining parts. This design illustrates how elastomeric bearings can simplify rotors.

The applications of elastomeric bearings in rotors were selected for investigation because of the severe oscillatory loads and motions that are present in this environment. These bearings should be considered, however, for other applications where oscillatory motions create problems. Figure 19 shows one such application: the small dot is the location of the bearing in Bell's Huey Cobra. This combination radial-thrust bearing was designed to support the collective-lever idler link. Endurance tests showed it to be satisfactory, so it became a production part. Early failures that affected the bearing formerly used have been eliminated.

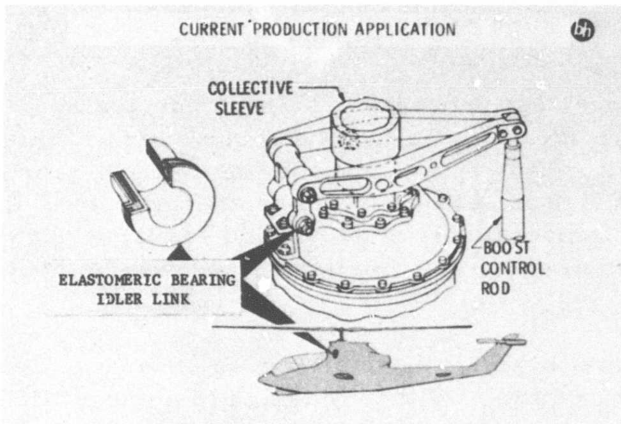

Figure 19

In one set of endurance tests, a thrust bearing in the "soft" tail-rotor configuration was put through 750 start-stops and 30,000 torsional cycles through 17 degrees on the tail-rotor whirl stand. Figure 20 shows its condition before and after the test program. The small amount of elastomer extruded demonstrates the fail-safe characteristic of the bearing. In fact, elastomer could be extruded to this extent all over, and the bearing would still carry the design loads. The amount of elastomer shown does not indicate that the bearing's service life is expended; the replacement conditions for elastomeric bearings are yet to be determined. The thrust bearings's spring force characteristics were measured before and after the test, and no change was noted.

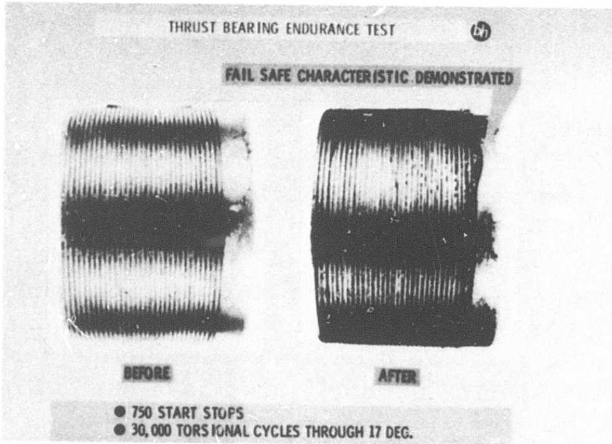

Figure 20

Another endurance-test program is being conducted at Lord Manufacturing Company under contract to Bell. One requirement of this program is a 900-hour test with loads and motions on the bearing representative of those that it would receive in service in the "soft" tail rotor. Testing will also be conducted at temperatures of -40°F and +140°F. This program is expected to furnish as much information as possible without a field-service evaluation.

The flapping radial bearings have also been endurance-tested, simulating a 2000-hour service life. With a redesign to enlarge the bearings slightly, their calculated service life can be extended to 6000 hours.

As Figure 21 shows, the suitability of elastomeric bearings for tail rotors has been demonstrated, but the rotor needs a service evaluation. The inplane-stiff main rotor has been analytically investigated. It appears to be promising. A research and development program should be initiated, however, if we are to realize all the benefits of elastomeric bearings in main rotors.

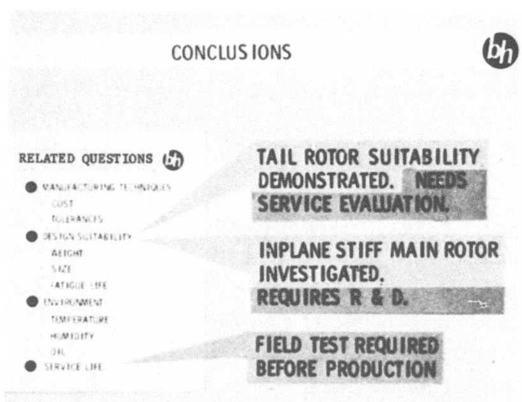

Figure 21

For either rotor, a field-test program before production is necessary. The tail-rotor service evaluation not only would verify the advantages of elastomeric bearings in two-bladed seesaw rotors, but would supply important design information for future rotors. Bell Helicopter Company recommends a tail-rotor service evaluation and a research program to evaluate the inplane-stiff main rotor.

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HISTORY, APPLICATION, AND SERVICE EXPERIENCE
WITH ELASTOMERIC BEARINGS OF THE
ENSTROM MODEL F-28A HELICOPTER

by

Paul L. Shultz
Enstrom Corporation
Division of the Purex Corp., Ltd.

SUMMARY

Although the Enstrom Corporation and its Model F-28 helicopter have gained general recognition in the helicopter industry, there is little general knowledge of the use of elastomeric bearings in its design and specifically that 7 years and approximately 10,000 hours of service experience have been accumulated. This paper documents the history of the F-28 helicopter, the design philosophy and details of application of elastomeric bearings to the main rotor system, and some comments on our experience with these bearings in commercial service.

HISTORY

In late 1960 and 1961, the Enstrom F-28 helicopter was designed. Through Hank Velkoff, then at Wright Field, we learned of an elastomeric pad bearing development by Hinks and Schook, and a decision was made to make this device a basic part of the main rotor hub design. Figure 1 shows this helicopter.

In September, 1961, a visit was made to Mr. Nelson Daniel of Fort Eustis to investigate the possibility of R&D funding for development of an all-elastomeric rotor hub design.

May, 1962, was the date of the first flight of the F-28 helicopter--the first flight of a helicopter with elastomeric bearings.

Figure 1

In 1963, the Hinks-Schook patent was acquired by the Marlin-Rockwell Corporation and first elastomeric pad bearings were received from this vendor under the trade name "LAMIFLEX".

In April, 1965, the Enstrom F-28 helicopter received FAA Type Certificate No. H1CE.

In August, 1965, U.S. Patent No. 3,200,887 covering application of elastomeric bearings to helicopter rotor systems was granted to the Enstrom Corporation.

In April, 1968, the Model T-28 helicopter was first flown, again with an elastomeric bearing rotor hub.

In May, 1968, the Model F-28A was certified.

In October, 1968, the Enstrom Corporation became part of the Purex Corporation.

APPLICATION TO MAIN ROTOR HUB

In applying the elastomeric bearing to the Model F-28 main rotor hub in 1961, the only form available at that time was the flat laminate pad suitable for thrust (or compressive) loads. We, therefore, elected to use conventional rolling element bearings for radial loads and to apply the laminated pad (in the form of an annulus) to C_F forces in the hub. Considerations in configuring the bearings were:

1. <u>The area required for C_F loads in the F-28 hub.</u>

   A design compressive value of 10,000 psi was used based on data developed by Hinks and Schook and later substantiated by Marlin-Rockwell.

2. <u>The torsional spring rate of the bearing.</u>

   Determination of propeller moment and mass polar moment of inertia of the total feathering system indicated that a torsional spring rate of 8 inch-pounds per degree for the elastomeric bearing would result in minimum cyclic stick loads. Thus the torsional spring characteristics of the bearing were utilized to benefit the dynamic characteristics of the feathering system.

   To attain the desired spring rate, the inside and outside diameter were kept as small as possible and the number of laminations required was determined. The resulting laminate stack then required stabilization, which was accomplished by incorporating stabilizing (or slipper) rings at intervals of .080".

Let us look at the anatomy of a "LAMIFLEX" bearing as furnished to Enstrom by the Marlin-Rockwell Corporation. Figure 2 shows such a bearing. In order to understand it better, let us take a section through the bearing and magnify it many times as shown in Figure 3.

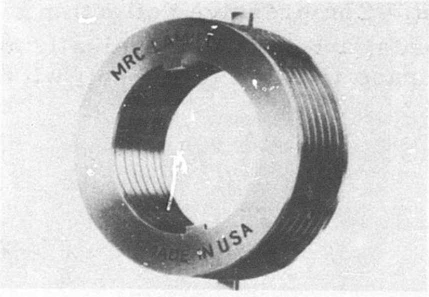


Figure 2

There are five laminate sections, each consisting of 23 to 24 metal laminations of .002" thickness and a corresponding number of rubber laminations of about .0015" to .0020" thickness. Each of these laminate sections is approximately .080" thick, which represents the optimum

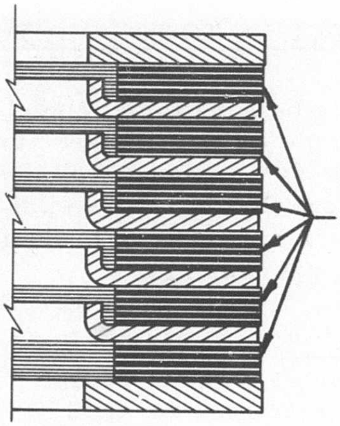


Figure 3

stack height for column stability under 10,000-psi loads. To attain the desired torsional spring rate of 8 inch-pounds per degree with the diameters we have established, it is necessary to have a total stack height of .510". To stabilize this stack as a column under the design compressive load, a series of stabilizing (or slipper) rings is added with a flanged inside diameter, which in turn bears on the Teflon-lined spindle. A .060"-thick ring at each end is keyed by tabs to the blade grip and the hub spindle.

Installation of the bearing in the main rotor hub feathering system is shown in Figure 4. The inboard end ring is keyed to the blade grip (or cuff), and the outboard end ring is keyed to the spindle. The bearing is set for zero torsional deflection with blade pitch at 6 degrees at the 3/4 radius. Compressive deflection of this bearing is .016" to .020" under C_F loads; and to prevent misalignment of the feathering bearings, the outer race is fitted to the cuff with more clearance than is customary.

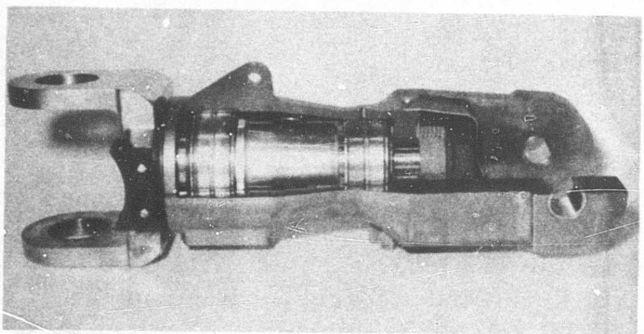


Figure 4

A further benefit of the elastomeric bearing is its use in providing a centering force for cyclic control, which is accomplished by increasing the torsional spring rate beyond that required to equate propeller moment to inertia forces.

Other considerations in developing the installation were (refer to Figure 4):

1. Location where small diameters relating to the low torsional spring rate were possible.

2. Location where bearings would not be exposed to lubricant provided for conventional bearings (note: O-ring used to restrain lube).

3. Easy access for removal or inspection without dismantling the hub.

Consideration has been given to the use of conical radial-type elastomeric bearings in the flapping and lag hinges of the rotor hub, and detailed design of this application was completed some time ago. So far, a decision to apply these bearings to production helicopters has been deferred because of the relating cost, which would be many times that of the rolling element bearing now used. Use of radial-type elastomeric bearings in the tail rotor and swash plate is under consideration.

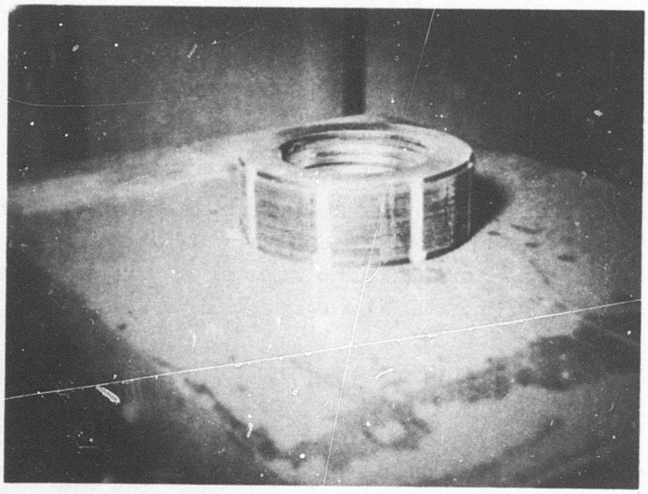
SERVICE EXPERIENCE

In the nine years that have elapsed since the F-28 main rotor hub design was established with elastomeric thrust bearings, some 10,000 hours of flight experience have been accumulated on about 30 helicopters, with good results. Operation in temperature ranges from -30°F to 120°F has been entirely successful with natural rubber bearings.

There have been two cases of bond or laminate separation and one partial metal laminate failure. Since the elastomeric bearing is in compression, a "fail-safe" condition exists. Drastic deterioration of the laminate would result only in very evident roughness of the blade feathering system.

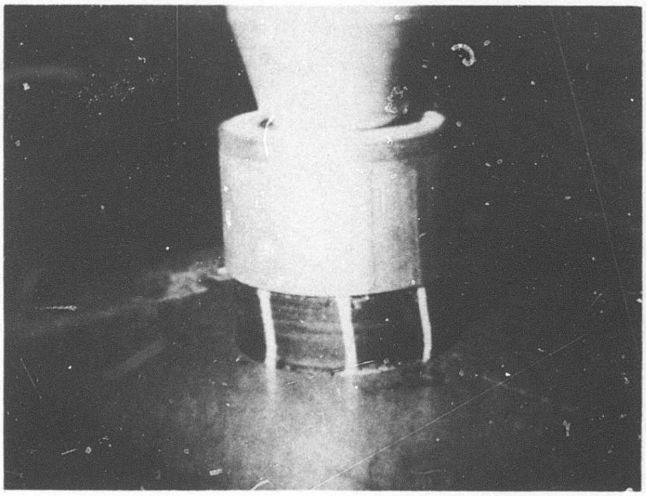
Recently, in attempting to attain ultimate rotor smoothness, we have been more critical of frictional forces which relate to the bearing of slipper rings of the elastomeric bearing on the Teflon lining of the spindle. We

have noticed evidence of heavy bearing of the slipper rings on the Teflon lining on the spindle which has been associated with mild rotor roughness and in some cases cyclic stick whirl. To determine if there was some variation in the column stability of the bearings being furnished to us and if this was a condition that had not existed earlier, we tested a number of bearings by applying a compressive load without stabilization of the bearing stack. The design C_F for the Model F-28A helicopter is 16,400 lbs. One of the oldest bearings available for test was supplied to us in 1963 and is shown unloaded in Figure 5 and under an 18,000-lb load in Figure 6. The straight lines painted on the side show that it is quite stable under this load. Figure 7 shows serial number 148, manufactured in 1966, under a 15,000-lb compressive load with some indication of instability. Figure 8 shows a bearing, manufactured in 1968, under a 13,000-lb load with noticeable instability. To isolate the effects of aging on the natural rubber laminations, lateral shear deflection tests were made on all bearings with consistent results, indicating that no apparent change in resilience could be attributed to age within a 5-year period. As a result of these tests, the width of the flange of the slipper rings has been increased to the maximum possible to reduce the unit bearing load on the spindle, and this has been quite helpful. It is planned to test a semiconical type of laminated pad bearing similar to that shown in Figure 9 in an effort to attain increased column stability; if it is as successful as we expect, we will incorporate this design in our helicopter.



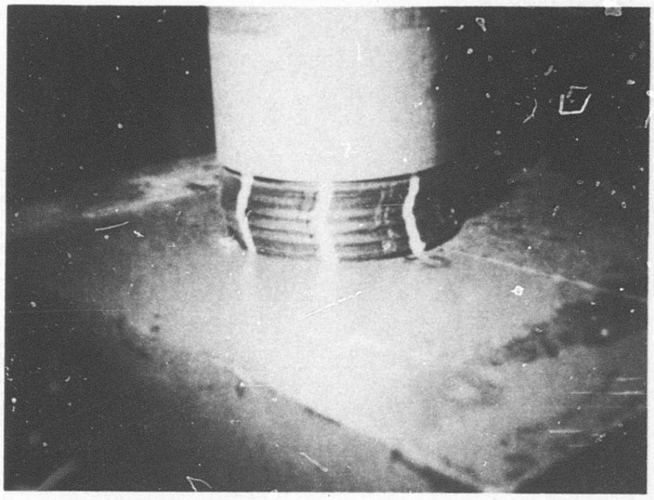
No Load

Figure 5



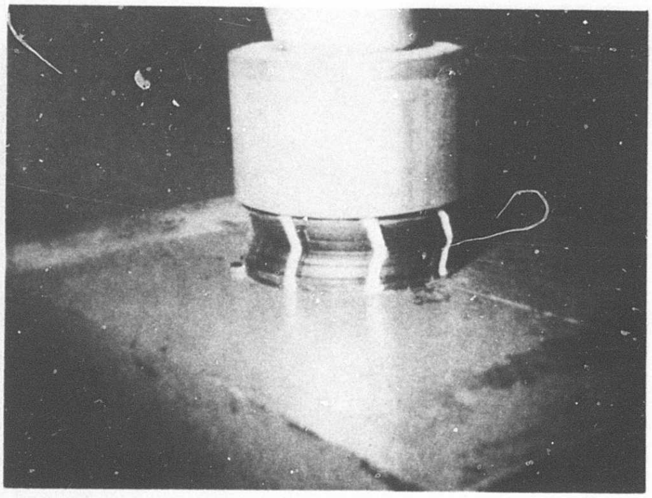
MRC (1963) - 18,000-lb Load

Figure 6



Serial No. 148 - 15,000-lb Load

Figure 7



Serial No. 75 - 13,000-lb Load

Figure 8

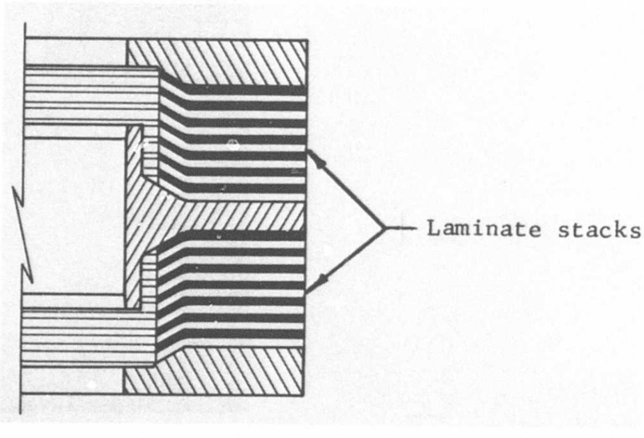


Figure 9

We have learned that the elastomeric bearing, when properly designed, is an item that, once installed, can be expected to provide trouble-free, maintenance-free service for many, many years; and it is our intention to adapt this unique device to other areas of our helicopter, such as the tail rotor and swash plate, where service requirements justify their cost. In addition, new helicopter models now being certified will also incorporate elastomeric bearings in the main rotor hub.

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13. ABSTRACT

This document is a compilation of the papers presented at the Elastomeric Bearing Symposium held at the US Army Aviation Materiel Laboratories, Fort Eustis, Virginia, 18 March 1969.

The objective of this symposium was to disseminate information generated to date on elastomeric bearings to aircraft airframe manufacturers as well as cognizant Government personnel.

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